1955

Investigation of failures of a high-speed crankshaft.

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Monterey, California: U.S. Naval Postgraduate School

http://hdl.handle.net/10945/14327

INVESTIGATION OF FAILURES OF A

HIGH-SPEED CRANKSHAFT

GEORGE BJORN LINDGREN

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OF A
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by

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Submitted in partial fulfillment
of the requirements
for the degree of
MASTER OF SCIENCE
IN
MECHANICAL ENGINEERING

United States Naval Postgraduate School
Monterey, California

1 9 5 5

This work is accepted as fulfilling

the thesis requirements for the degree of

MASTER OF SCIENCE

IN

MECHANICAL ENGINEERING

from the

United States Naval Postgraduate School

PREFACE

In the Indianapolis Classic of 1954 every race car entered employed the same type engine of Meyer-Drake extraction. There has been no basic change in this engine since it was first received by the motor racing world in 1922.

In recent years, due to the application of higher compression ratios and more powerful fuels, the Meyer-Drake crankshafts have been experiencing a great number of failures. Many crankshafts, upon the completion of a race, have been found with cracks initiated in them, indicating that the shafts were operating close to their endurance limits.

The author wishes to express his appreciation for the advice and suggestions of Professor A. K. Schleicher of the Mechanical Engineering Staff of the U. S. Naval Postgraduate School. His guidance and painstaking check of the course of action in detail aided greatly in this investigation.

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TABLE OF SYMBOLS AND ABBREVIATIONS

(Listed in the order of their use in the text)

F_c	-	Inertia Force for the Revolving Crank Mass
M_c	-	Mass, Crank
R	-	Radius of Crank
ω	-	Angular Velocity, radians/sec
F'_p	-	Inertia Force for the Reciprocating Mass(Primary)
M_p	-	Mass, Reciprocating
θ	-	Crank Angle Measured Counter-Clockwise from Top Center
F''_p	-	Inertia Force for the Reciprocating Mass(Secondary)
L	-	Length of Connecting Rod
F_b	-	Inertia Force for the Brass Counterbalance Crescents
M_b	-	Mass, Brass Counterbalance Crescents
F_s	-	Inertia Force due to the Removal of Segments From Cheeks
F_h	-	Inertia Force due to Increasing the Diameter of the Pinbore

CHAPTER I

INTRODUCTION

This investigation was conducted to improve the durability of the crankshaft presently installed in the Meyer-Drake (Or Offenhauser) engine. This four cylinder, 270 cubic inch displacement engine, employing compression ratios up to 14.5 to one, develops from 405 to 410 brake horsepower at 6,000 rpm, operating on a fuel mixture of ten percent nitro methane added to methanol.

The engine is constructed of three major components - the head, the cylinder block, and the crankcase. Tie-rods running from the cylinder head to the crankcase had to be added to prevent separation of the cylinder block and crankcase during high-speed operation.

There has been an increasing number of crankshaft failures forward of the center main bearing and often occurring at the forward fillet of the crankpin on the foremost throw. The failure usually manifests itself as a crack running along the crankpin fillet perpendicular to the axis of symetry of the shaft such that the midpoint of the crack is uppermost when the throw is at bottom center.

An interesting fact to note is that the Meyer-Drake engine was not plagued by crankshaft failures to any significant degree before the advent of nitro methane fuel.

There are several possible solutions to the problem of increasing the durability of the shaft. These are to reduce the forces imposed on the crankshaft through proper balancing and to increase the strength of the shaft.

It is standard practice to provide balance for all the revolving crank mass, including that part of the mass of the connecting rod assumed concentrated at the crankpin, and from one-half to two-thirds of the reciprocating mass. The effect of using a counter-balancing weight equal to the sum of the crank weight and the reciprocating weight would neutralize all the vertical forces except the secondary, but would result in quite large horizontal forces instead.

The balance is not proper in the present engine. It is felt by some of those cognizant of the engine's operating conditions that the failure is initiated when the car is decelerated for entering one of the curves. When the driver closes the throttle the engine is often operating at 6,000 rpm, and thus the full effect of the unbalance forces is encountered.

The crankcase housing, the crankshaft, and its five main bearings constitute a separate assembly of so-called barrel construction with the bearings mounted in spoked disks which are in turn bolted to the crankcase housing.

The strength of the shaft may be increased by improving the material, changing the heat-treatment, altering the shape, or by any combination of these three factors.

The crankshaft is now made from S.A.E. 6145 chromium-vanadium steel hardened to a hardness of 36-38 on the Rockwell C scale with no special surface treatment.

One method to improve the crankshaft by changing the heat treatment would be to nitride the section. Investigations have shown that nitriding may increase the fatigue strength of the part by as much as twenty percent. Proper heat treatment and perhaps nitriding the shaft would be of definite merit. However, heat treatment is beyond the scope of this investigation. ("Fatigue of Metals" by R. Cazaud[1] and "The High-Speed Internal-Combustion Engine" by H. R. Ricardo[2].)

Finally the durability of the crankshaft may be improved by redesigning the shaft so as to provide better stress distribution. It would be desirable to maintain the principal over-all dimensions of the crankshaft so that a major engine redesign could be avoided.

SUMMARY

It is the purpose of this work to accomplish the following:

a. determine cause of crankshaft failure,

b. provide more satisfactory balance of the engine,

c. provide better stress distribution through redesigning the shaft.

CHAPTER II

BALANCING OF CRANKSHAFT

The crankshaft has counterweights on each crank throw; thus an attempt has been made to balance each throw of the engine. As a result a minimum of force due to unbalance is transmitted through the crank from one throw to the next.

After the crankshaft has been machined, it is put on a balancing machine and the remaining unbalance in the crankshaft as a whole due to machining imperfections is removed. For this reason a good indication of the unbalance effects may be obtained by the investigation of one throw by itself. Therefore, one throw was taken and treated like a single cylinder engine. This gives comparative data as well as a good indication of the forces the crankshaft transmits due to unbalance. No attempt is made by the factory to provide for balance other than that of the crankshaft and that part of the connecting rod assumed part of the revolving mass.

Due to the small clearance between the cheeks and the connecting rod, a counterbalance mass of a maximum thickness of only five-sixteenths inches may be added to each cheek.

MEASUREMENTS

Length of throw:	$2\frac{1}{4}$ inches
Length of connecting rod:	8 inches
Weight of connecting rod:	1,493 gms.
Weight of connecting rod assumed concentrated at wristpin:	392 gms.
Weight of connecting rod assumed concentrated at crankpin:	1,101 gms.
Weight of piston and rings:	895 gms.
Weight of wristpin:	302 gms.
Total Reciprocating weight:	1,589 gms.

Converting the weights into pounds, the total inertia forces are as follows for a speed of 6,000 rpm:

For the revolving crank mass,

$$F_c = M_c R\omega^2$$

$$= \left(\frac{2.43}{32.2}\right)\left(\frac{2.25}{12}\right)\left(\frac{6000 \times 2\pi}{60}\right)^2 = 5,600 \text{ lbs.}$$

For the reciprocating mass (primary),

$$F'_p = M_p R\omega^2 \cos\theta$$

$$= \left(\frac{3.5}{32.2}\right)\left(\frac{2.25}{12}\right)\left(\frac{6000 \times 2\pi}{60}\right)^2 \cos\theta = 8,050 \cos\theta \text{ lbs.}$$

For the reciprocating mass (secondary),

$$F''_p = M_p \left(\frac{R^2}{4L}\right)\left(2\omega^2\right) \cos 2\theta$$

$$= \frac{R}{L} F'_p \frac{\cos 2\theta}{\cos \theta}$$

$$= \left(\frac{2.25}{8}\right)\left(8{,}050\right) \cos 2\theta = 2{,}265 \cos 2\theta \text{ lbs.}$$

For the brass counterbalance crescents, each weigh 1.45 pounds with a center of gravity two and one-third inches from the main axis of the shaft, or the equivalent counterweight is 1.31 lbs. located with a center of gravity two and one-fourth inches from the main axis of the shaft.

$$F_b = M_b R^{\ 2}$$

$$= \left(\frac{2 \times 1.31}{32.2}\right)\left(\frac{2.25}{12}\right)\left(\frac{6000 \times 2\pi}{60}\right)^2$$

$$= 6{,}030 \text{ lbs.}$$

To provide balance for all of the revolving mass and one-half of the reciprocating mass, there must be a total of 4.18 lbs. weight provided. This produces an inertia force of,

$$F_b = \left(\frac{4.18}{32.2}\right)\left(\frac{2.25}{12}\right)\left(\frac{6000 \times 2\pi}{60}\right)^2$$

$$= 9{,}630 \text{ lbs.}$$

Since it would be rather difficult to add more metal to the counterbalance lobes because of the small clearances involved, and since metals heavier than brass are unfortunately either too soft or too expensive, other

schemes must be undertaken to obtain the desired balance.

One method which may be used to achieve a more favorable balance is to turn on a lathe or mill off a segment of the cheeks on the outside of the upper portion of the throws. The former method was adopted because a slightly larger amount of metal could be removed in this manner for the same amount of cheek thickness as measured on the top of the throw.

It was decided to turn the shaft about its main center and start taking a cut at the radius of the throw, two and one-fourth inches, and that a cheek thickness of eleven thirty-seconds inches should be left remaining at the top of the throw. This design is a result of a survey of data on previous designs and tests. ("Fatigue of Metals" by R. Cazaud[1], "The High-Speed Internal-Combustion Engine" by H. R. Ricardo[2], "Experimental Stress Analysis", Vol. II, Number 2 published by The Society for Experimental Stress Analysis[3].)

The complex three-dimensional shape of the segment to be removed made the process of figuring its weight and center of gravity rather difficult. This problem was solved by making a full size mold of the segment, filling it with water, and thus obtaining its volume. A model of the crescent-shaped wedge was then cast in paraffin. By supporting the wax segment by threads from various points and using a plumb line, the center of gravity was found.

The values obtained are listed below:

Volume of segment: 15cc of water = 0.915 cu. in.

For two segments this represents a weight of steel of

$$2 \times .915 \times .283 = .52 \text{ lbs.}$$

The center of gravity of the segment was found to exist at a radius of two and thirteen-sixteenths inches from the main axis of the shaft.

The effect of removing material from the throw is equivalent to adding an equal amount of weight at the same radius on the counterbalance lobes. Therefore, the equivalent counterweight at a radius of two and one-fourth inches is

$$0.52 \times \frac{2.8125}{2.25} = 0.65 \text{ lbs.}$$

The inertia force of this additional counterbalance is,

$$F_s = \left(\frac{.65}{32.2}\right)\left(\frac{2.25}{12}\right)\left(\frac{6000 \times 2\pi}{60}\right)^2 = 1{,}500 \text{ lbs.}$$

At present the lightening hole for both the main journal and the crankpin has a diameter of nine-sixteenths of an inch. This diameter is smaller than any found in comparable engines. Before increasing the bore of the lightening holes, the effect stress-wise of such a move was investigated.

Previous investigations show that the critical region for a crank throw in torsion generally occurs in the crankpin fillets somewhat below the center line of the throw with the principal planes oriented at about forty-

five degrees with the fillet as shown at A in Fig. 1. A similar critical region exists in the main journal fillets, except that due to the heavier construction normally employed in the journals, these regions are usually less dangerous. When the crank throw is left solid, high values of stress are confined to these regions.

If the crankpins and journals are bored to lighten the shaft a partial collapse of the ends of the pins and journals occur as shown in Fig. 2. Concentrations of tensile stress occur at B and C, and compressive stress occur at D and C, with the principal stress oriented parallel to the edge of the lightening hole. Stresses are greater at B than at C, and are greater in the pin than in the journal if heavier sections are employed in the latter.

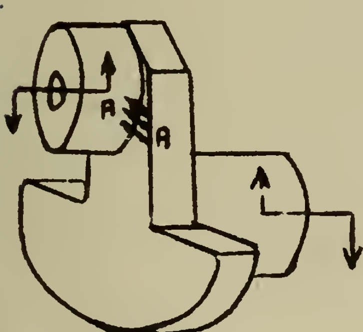


Fig. 1 Location of Critical Torsional Stress

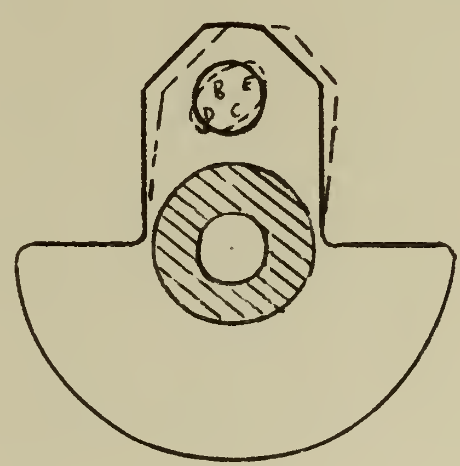


Fig. 2 Distortion of Hollow Crankpin Under Torsion

In the case of hollow shafts, although the stresses in the fillets are more dangerous than at the edge of the lightening hole, it is possible, if the hole is made too large, to obtain stresses that are more severe at the edge of the hole than in the fillet.

With respect to bending stresses, only distortion of the type encountered when resonant bending vibration is present, or when an overhung load exerts a reversed bending couple at the end of the shaft is considered. Greatest deflection and highest stresses have been found to occur when the bending moment is applied in the plane of the throw.

As shown in Fig. 3, the abrupt change in cross section at the journals and pin fillets, at sections FF and GG, resulted in the highest concentrations of stress there.

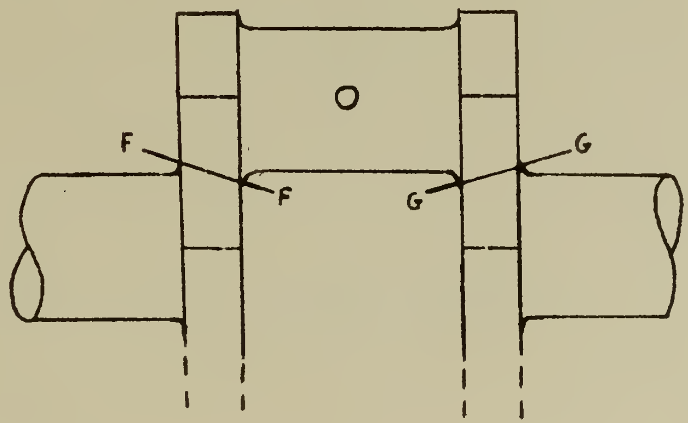


Fig. 3 Points of Highest Concentrations of Stress due to Bending

The nature of these concentrations depend in a large measure upon whether or not lightening holes are present. If journals and pins are solid, stress reaches a maximum at the center, point H, Fig. 4; but if they are bored the stress at the center recedes and a pair of peaks JJ are formed.

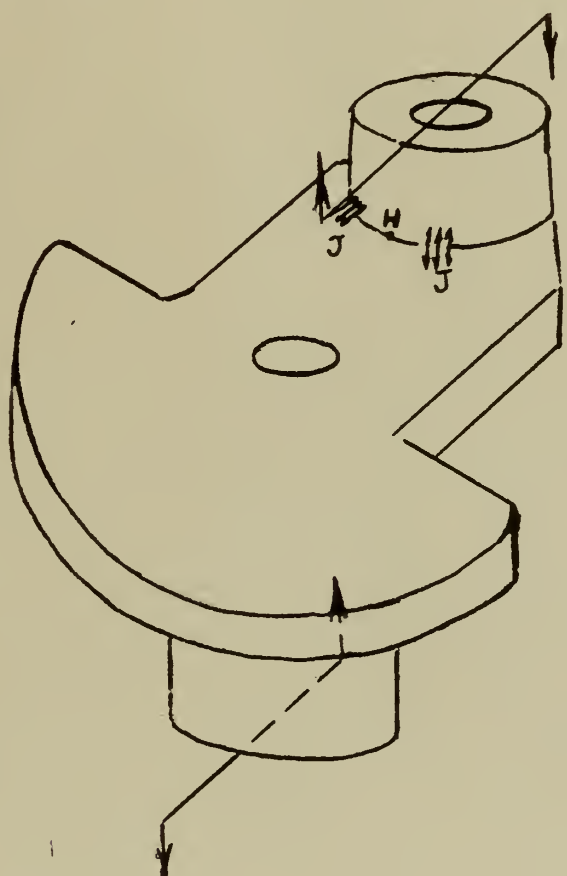


Fig. 4 Location of Critical Bending Stresses

In both bending and torsion it is possible to use an optimum lightening-hole size, such that a deviation either way aggrevates the fillet stress. From curves of previous investigations it is shown that using a ratio of pinbore to pin diameter of from about five-tenths to six-tenths resulted in optimum conditions.

In order to take advantage of the resulting better balance, a ratio of pinbore to pin diameter of six-tenths was decided upon. ("Fatigue of Metals" by R. Cazaud[1], "The High-Speed Internal-Combustion Engine" by H. R. Ricardo[2], "Experimental Stress Analysis", Vol. II, No. 2, published by The Society for Experimental Stress Analysis[3], and "A Short-Gage-Length Extensometer and Its Application to the Study of Crankshaft Stresses" by C. W. Gadd and T. C. Van Degrift[4].)

The volume of the crankpin lightening hole on the original crankshaft is 0.69 cubic inches. (This is the volume of only the void - the plugs are left in place.)

The crankpin has an outside diameter of 2.123 inches. Therefore, the new pinbore will have a diameter of 0.6 x 2.123 = 1.274 inches, or a volume of 3.55 cubic inches.

$$V_{\text{pinbore proposed}} - V_{\text{present pinbore}} = 3.55 - .69 = 2.86 \text{ cubic inches}$$

A volume of 2.86 cubic inches represents a weight of .283 x 2.86 = 0.81 lbs.

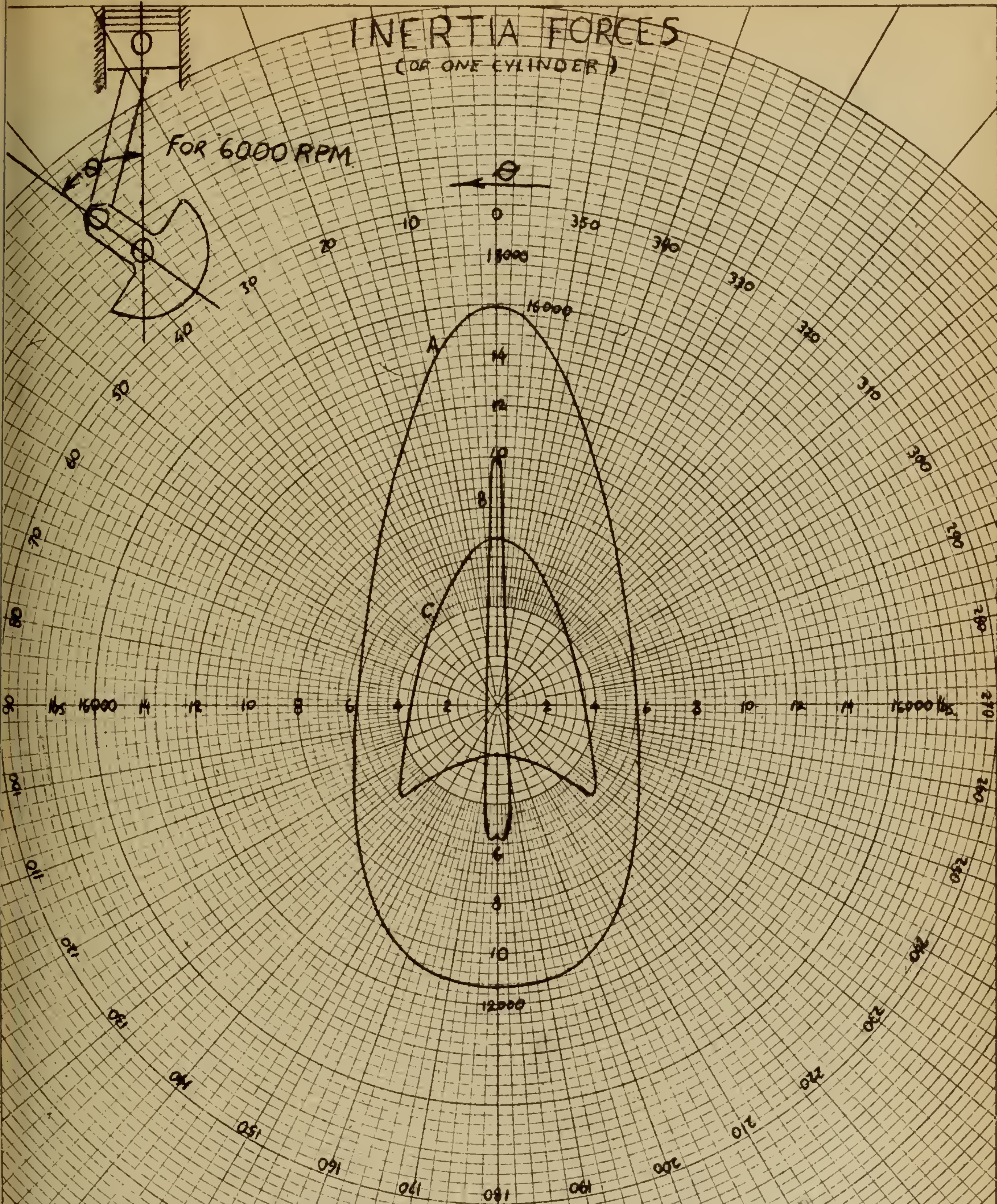
At 6,000 rpm this represents an inertia force of:

$$F_h = \left(\frac{.81}{32.2}\right)\left(\frac{2.25}{12}\right)\left(\frac{6000 \times 2\pi}{60}\right)^2$$

$$= 1{,}865 \text{ lbs.}$$

By this scheme we have obtained a total inertia force at 6,000 rpm equal to the sum of $F_b + F_s + F_h$ or $6{,}030 + 1{,}500 + 1{,}865$ or 9,395 pounds. Thus, practically all of the rotating mass and one-half of the reciprocating mass has been balanced. Curve C of Fig. 5 shows the resulting inertia forces of one throw of the engine in this state of balance.

The additional weight required to obtain a balance of over one-half of the reciprocating mass can easily be obtained by boring out the cénter of the counterbalance lobes and filling the void with brass, which has a specific weight of .308 lb/cu.in. versus .283 lb/cu.in. for that of the steel.



CURVE A: BALANCE LOBES ONLY.
CURVE B: COUNTERWEIGHT NOW ON CRANKSHAFT (1.45 lbs/cheek).
CURVE C: RESULT FROM BORING A 1.274 DIAM. LIGHTENING HOLE IN PIN AND REMOVING A 0.26 lb SEGMENT FROM EACH CHEEK (SEE DETAILS IN TEXT).

FIG. 5

CHAPTER III

REDESIGN OF THE CRANKSHAFT

Studying the original crankshaft the following poor design features were noted:

(a) Crankshaft does not permit proper balance of engine as noted in the previous chapter.

(b) All fillets have abnormally small radii for the general proportions involved.

(c) There was no overlap of the crankpin with the journal.

One particularly favorable design feature noted was that the oil-hole in the crankpin was placed on the neutral plane of the crankshaft (at 90 degrees in the direction of rotation). There has been no record of failures originating at the oil-hole on the Meyer-Drake crankshaft.

An attempt to improve the poor features of the design was made in the following way:

(a) As discussed in Chapter II, it was decided that the lightening holes of the pin should be bored out to a value of 1.274 inches, and that the journal should be bored to a value of six-tenths of its outside diameter of 2.374 inches or 1.424 inches. This will allow better balance and when incorporated with other changes should

not increase the maximum stress to any great extent in the fillets.

(b) The zone between two parts of a shaft having different diameters should always be given as large a fillet radius as possible. Practically, it may be assumed that if the radius of the fillet is not less than half of the smaller diameter, the reduction in fatigue limit will not be greater than 15 percent. ("Fatigue of Metals", by R. Cazaud[1] and "The High-Speed Internal-Combustion Engine", by H. R. Ricardo[2].) It would be quite impractical to have a fillet radius to pin diameter ratio of one-half on a structure such as a crankshaft, but it would have been very desirable to increase the fillet radius by at least twice its present value, and thus decrease the stress concentration at that point; however, very limited space was available due to space that had to be occupied by the connecting rod bearing. It was thus decided to make the fillet with curved portions having different radii of curvature as shown in Fig. 6.

(c) The advantage of having overlap of the crankpin with the journal to some degree was investigated. Overlap to a certain extent has proven beneficial in crankshafts, increasing the stiffness and reducing the deviation of equi-potential lines of stress as they enter the web. (Figure 7 is reproduced from "The Fatigue

of Metals" by R. Cazaud[1].)

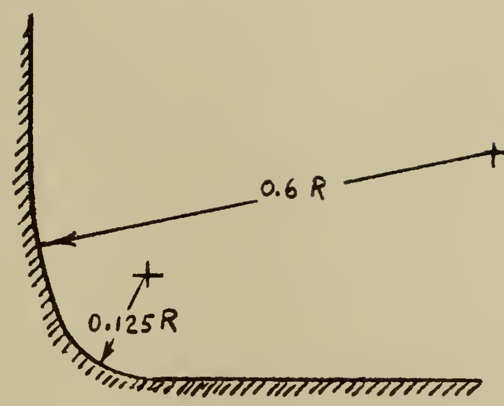


Fig. 6 Detail of Double Fillet

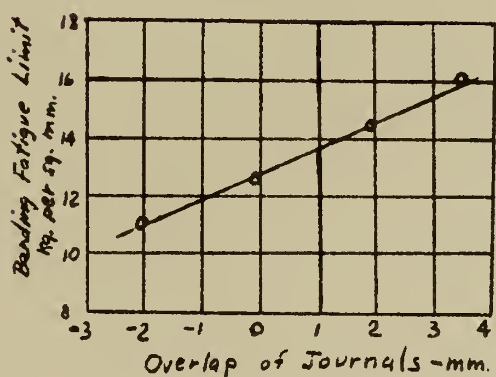


Fig. 7 Effect of overlap on the Fatigue Strength of a Crankshaft

From the design aspects of the original engine it was determined that the main journals could be increased in diameter from 2.374 inches to 2.624 inches, thus providing an overlap of one-eighth of an inch.

A test section consisting of a single throw of this new crankshaft was manufactured and a one-throw test section was cut out from the original crankshaft. A means of obtaining a comparison of the two designs was now desired.

CHAPTER IV

TESTING CONSIDERATIONS AND DESCRIPTION OF APPARATUS

It would have been desirable to have been able to place strain gages on cheeks of the crankshaft when it is installed in the engine, run the engine at 6,000 rpm, and obtain values of stress under dynamic conditions. Then the various pistons could have been loaded hydraulically at different crank angles using the engine as a supporting jig for the shaft until comparable stress readings under static conditions were found. The strain distribution on the shaft could then be reasonably determined.

Along this vein of thought, it would also have been very desirable to have available an extensometer with a gage length of one-sixteenth inch or less.

Since there were no provisions for testing dynamically or obtaining a short gage length extensometer, it was decided to test the two different crankshaft test sections with brittle lacquer and try to obtain a corrolation of strain values by applying SR-4 Type A-7 strain gages wherever it would be profitable and the section afforded the required space at a fairly constant strain level.

It was resolved to test the two designs in torsion and bending. For the torsion tests it was decided that the crank section was to be chucked into the Tinius

Olsen Torsion Machine by its main journals because it was the simplest form of loading which approaches torsional loading in the actual engine. (Fig. 8) It is recognized that due to the offset of the throw a complex loading results. For the bending tests, a jig was designed to provide as nearly pure bending as possible by providing a couple at the neutral axis of the main journals. (Fig. 9) Hardened inserts were provided for the knife edges to rest upon, thereby reducing any tensile forces produced to a minimum due to digging in of the supports. The jig was designed with a distance of six inches between the top knife edges and the bottom knife edges on each side. The desired bending moment was obtained through the application of load to the center of the top T-section in the Riehle 260,000 lb. Universal Testing Machine.

Fig. 8
Torsional Loading Arrangement for Crankshaft Study

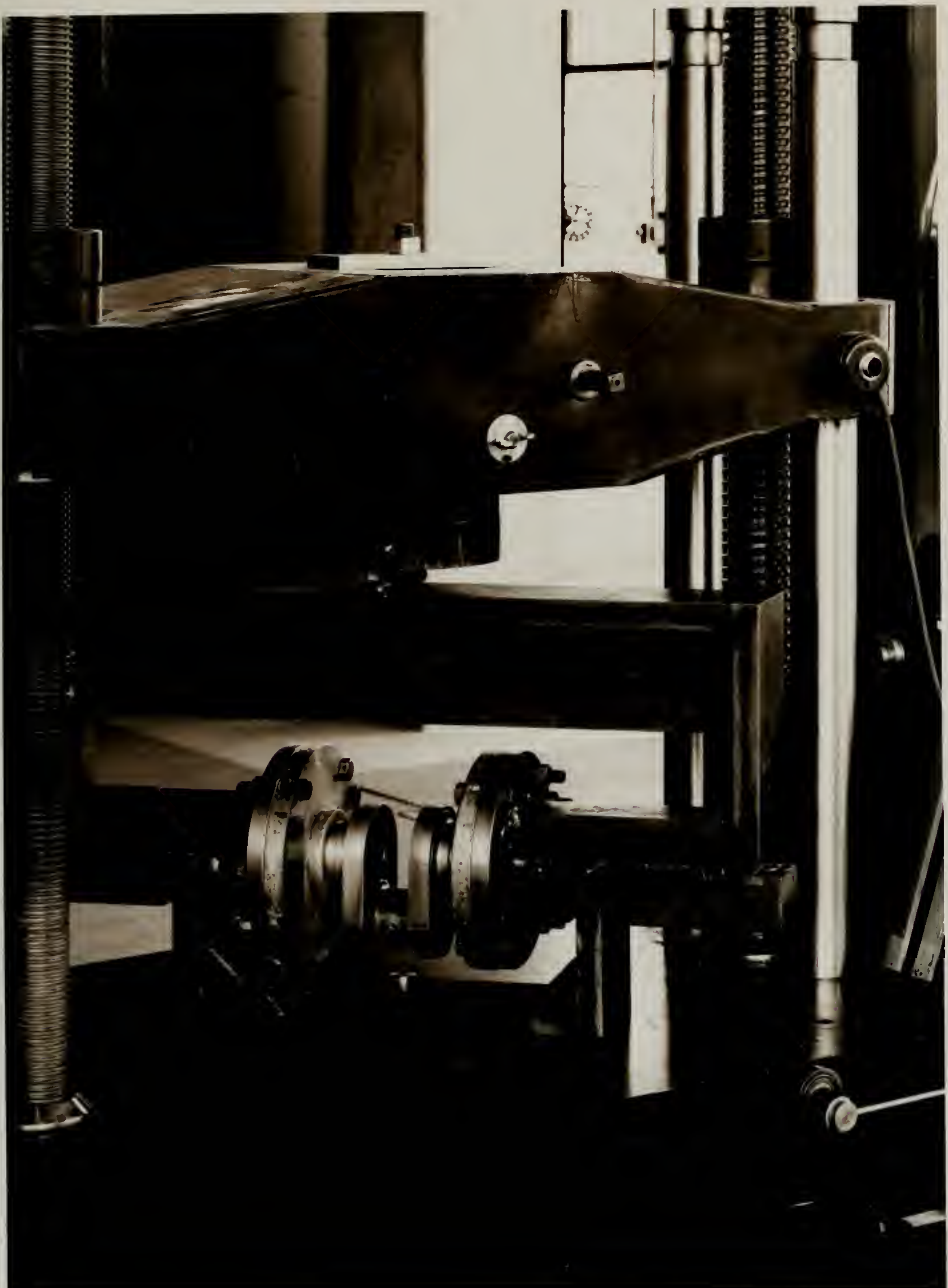


Fig. 9
Bending Jig for Crankshaft Stresscoat Study

CHAPTER V

TEST PROCEDURES

The following procedure for the preparation and testing of specimen was followed in the order named:

(a) The crankshaft test section and a number of calibration strips were sprayed with a Stresscoat lacquer coating calculated to provide a sensitivity in the order of .0008 inches/inch. Particular pains were taken to attempt to maintain an even thickness of lacquer throughout the crank section.

(b) The crankshaft test section and the calibration strips were dried for approximately one day in a drying box held at an average temperature of 120 degrees F.

(c) The wet and dry bulb temperatures were taken at the vicinity of the testing machine to be used, and the temperature in the region of the machine was regulated to maintain the proper sensitivity of the lacquer.

(d) The crankshaft section and calibration strips were allowed to cool to the ambient air temperature at the testing machine for a period of approximately eight hours.

(e) At the start of the test a calibration strip was loaded in the calibrator in one second or less, and the threshold sensitivity marked. The load was

removed, the calibration strip put in the strain scale, and the value recorded.

(f) Loading at a steady rate was commenced and the test section examined for cracks. When the first crack appeared, the loading was stopped and the time from the start of the test recorded. The cracked area was marked on the specimen.

(g) The test section was then loaded at the same rate as previously for an increment of 20-25 percent of the load attained upon the formation of the first crack(s).

(h) The specimen was then examined for the formation of the second crack(s), and the total time recorded. This procedure, thus formulated, was continued for the remainder of the test.

(i) At opportune moments during the test and at the completion of the test, calibration strips were loaded and checked for the threshold strain value.

(j) The actual value of strain present was derived from the Creep Correction Chart in "Stresscoat Operating Instructions", [5].

The brittle coating fractures perpendicular to the principal tensile strain. As the load is increased on the test section, those local areas which are the most highly strained in tension will be the first to form patterns.

The brittle lacquer is essentially indicating loads at which the same value of strain occurs on different areas of the structure in this kind of test. In order to compute all strains to any certain load, two conditions must be fulfilled. The material of the section must remain in its elastic range, and the distribution of loading must not change. With these two conditions fulfilled, it is sound to assume that all local strains vary directly in proportion with the load.

On appropriate tests, the directions of maximum principal strains as indicated by Stresscoat were scribed, and SR-4 wire strain gages were attached at points where the strain gradients were small. The A-7 gage which combines good accuracy with a short gage length of one-fourth inch was chosen.

CHAPTER VI

EXPERIMENTAL RESULTS

From the original test section it was determined that the failure of the crankshaft was due to bending and not to torsion. In the bending tests the brittle lacquer would produce a crack that was identical in position and appearance to that of the failed shafts. (Fig. 10)

1. Torsion Tests

Comparing results obtained by using Stresscoat in the original test section against SR-4 strain gage readings the following information is obtained:

Torque in.lbs.	Strain Indicated by Stresscoat, corrected for Creep	Strain Indicated by Corresponding SR-4 Strain Gages
56,000	.00114	.00131
60,000	.00116	.00104

For the strains at a torque of 56,000 in.lbs. this represents an error of:

$$\frac{.00131 - .00114}{.00131} = \frac{.00017}{.00131} = .13 \text{ or } 13\% \text{ error}$$

and for strains at a torque of 60,000 in.lbs., this shows an error of:

$$\frac{.00116 - .00104}{.00104} = \frac{.00012}{.00104} = .115 \text{ or } 11.5\% \text{ error}$$

(See Fig. 11)

Fig. 10
Brittle Lacquer Cracks due to Bending Loading

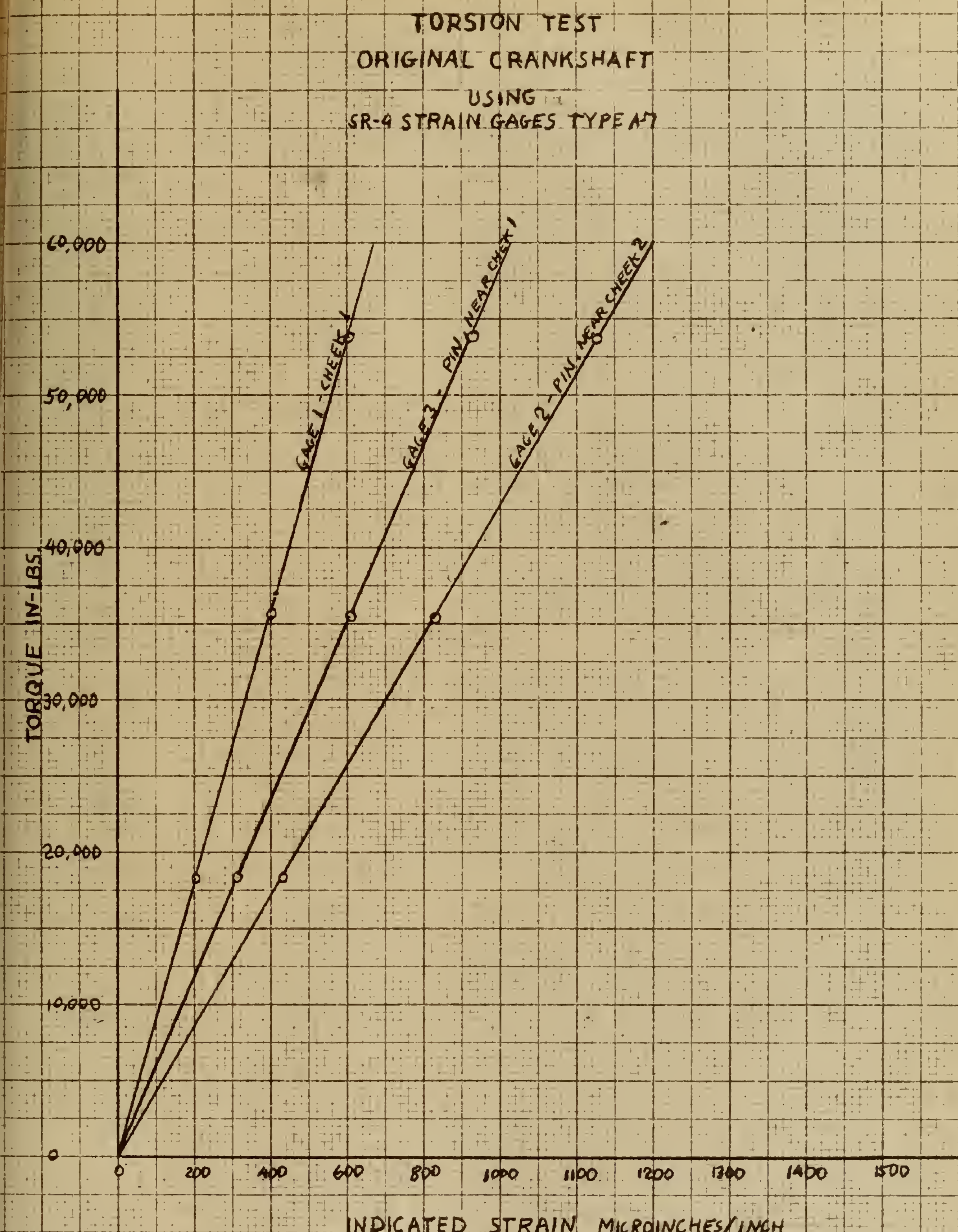

TORSION TEST
ORIGINAL CRANKSHAFT
USING
SR-4 STRAIN GAGES TYPE A7
GAGE 1 - CHEEK 1
GAGE 3 - PIN, NEAR CHEEK 1
GAGE 2 - PIN, NEAR CHEEK 2
TORQUE IN-LBS.
60,000
50,000
40,000
30,000
20,000
10,000
0
0
200
400
600
800
1000
1100
1200
1300
1400
1500
INDICATED STRAIN, MICROINCHES/INCH

Fig. 11

This is well within the expected accuracy of the lacquer. The effect of transverse strains was neglected. The A-7 gage has a transverse sensitivity factor of minus one percent.

Where there is present a rather high strain gradient, an average value for torque and indicated strains is chosen for comparison of data.

For convenience, all strains will be referred to 60,000 in.lbs. torque. In the original section, the first crack appeared in the fillet at an indicated strain of .00088 in/in for a torque of 23,500 in.lbs. or at 60,000 in.lbs. $.00088 \times \frac{60,000}{23,500} = .00225$ in/in apparent strain.

General cracking out in the pin took place at a torque of 60,000 in.lbs. for an apparent strain of .00116 in/in.

The strain ratio would therefore be

$$\frac{.00225}{.00116} = 1.94$$

In the redesigned crankshaft section, the first crack in the fillet appeared at a torque of 29,000 in.lbs. and at an indicated strain of .0098 in/in, therefore giving $.00098 \times \frac{60,000}{29,000} = .00203$ in/in apparent strain at 60,000 in.lbs.

General cracking took place out in the pin at a torque of 59,000 in.lbs. with an apparent strain value of .00121 in/in.

Thus, at 60,000 in.lbs. torque there would be an apparent strain of

$$.00121 \times \frac{60,000}{59,000} = .00123 \text{ in/in.}$$

For the same sensitivity, the maximum strains occurred at a torque of 60,000 in.lbs. for the original test section versus 59,000 in.lbs. for the redesigned crank section, producing comparative cracking out in the pin for both sections. Therefore, the maximum strain level was not raised in redesigning the crankshaft.

The strain ratio factor would be

$$\frac{.00203}{.00123} = 1.65$$

The shaft appears to be improved in torsion by a value of

$$\frac{1.94 - 1.65}{1.94} = \frac{.29}{1.94} = .15 \text{ or } 15\%$$

In the torsional tests the complex loading results in the crankpin not being parallel to the main axis of the shaft. In torsion the minimum principal stress is of opposite sign to the maximum principal stress. Since the maximum strain level was not raised in the redesigned crank section, it may be assumed that the

principal stresses were not significantly altered in magnitude and that the redesigning of the crank section did not result in a decrease in torsional durability.

2. Bending Tests

For elements loaded so that the point of maximum strain and the point of maximum distortion energy coincide and the loading is not far removed from the uniaxial system, the distortion energy has shown (even under fatigue conditions where the stress gradient is not abnormally high) that equivalent stress at the point of maximum energy correlates very well with the stress at the endurance limit of the material.

In crankshaft bending tests it has been shown there is little difference between the maximum strain theory of strength and the distortion energy theory of strength. ("Stress Concentration and the Fatigue Strength of Engine Components" by C. Gadd, N. A. Ochiltree, and A. Zmuda, published in the "Symposium on Testing of Parts and Assemblies" June 26, 1946 by the American Society for Testing Materials[6].)

All bending tests were run at a Stresscoat sensitivity of .0008 in/in. For the comparison of bending test data, all strains will be referred to 5,000 lbs load (1250 ft.lb. bending moment.)

In the original section, the first crack appeared in the fillet at an indicated strain of .00095 in/in at a load of 1,200 lbs, or at 5,000 lbs load the apparent strain is:

$$.00095 \times \frac{5000}{1200} = .00396 \text{ in/in.}$$

Cracking in the pin at a distance of .43 inches from the cheek at a load of 6,000 lbs for an apparent strain of .00129 in/in, which at 5,000 lbs load is:

$$.001295 \times \frac{5000}{6000} = .00108 \text{ in/in.}$$

This results in a strain ratio of:

$$\frac{.00396}{.00108} = 3.66.$$

Cracking in the cheek occurred at an average load of 5,500 lbs for an apparent strain of .00127 in/in. At 5,000 lbs load this is an apparent strain of:

$$.00127 \times \frac{5000}{5500} = .00115 \text{ in/in which}$$

gives a strain ratio of

$$\frac{.00396}{.00115} = 3.44.$$

In the redesigned crankshaft section, the first fillet crack appeared at a load of 1,145 lbs for an apparent strain of .00089 in/in. This represents at 5,000 lbs load an apparent strain of:

$$.00089 \times \frac{5000}{1145} = .00389 \text{ in/in.}$$

Cracking in the pin at a distance of .42 inches from the cheek took place at a load of 5,410 lbs for an apparent strain of .00124 in/in, or at a load of 5,000 lbs. This represents:

$$.00124 \times \frac{5000}{5410} = .00115 \text{ in/in.}$$

The strain ratio is thus:

$$\frac{.00389}{.00115} = 3.38.$$

At a load of 5,410 lbs, cracking took place in the same area of the cheek as on the original test section at an apparent strain of .00124 in/in.

At 5,000 lbs load this again represents an apparent strain of .00115 in/in, which also results in a strain ratio of 3.38.

TABULATED RESULTS & APPARENT STRAINS
(based on a load of 5,000 lbs)

Position	Original Crank section	Redesigned Crank section
Fillet(1st crack)	.00396	.00389
Pin, at approx. .42 in. from cheek	.00108	.00115
Cheek, area parallel to 180° position of pin fillet	.00115	.00115

TABULATED STRAIN RATIOS

References	Original Crank section	Redesigned Crank section
fillet/pin	3.66	3.38
fillet/cheek	3.44	3.38

The differences in the values obtained both in torsion and bending are insignificant because they all fall within the accuracy obtainable by Stresscoat.

From the apparent strain values for the cracks appearing in the crankpin, it may be that the indicated lower strain ratio obtained for the new design results from the raising of the strain level in the pin.

It was decided to run an additional bending test in the following manner. Apply load to the testing apparatus until the threshold crack in the fillet appeared, record the load and time, then continue loading at a steady rate until cracks in the pin occurred. This test would provide a good check on the creep correction

employed for the loading times involved. The following result was obtained.

PREVIOUS BENDING TEST

Load	Crack	Total Time	Creep Corrected Apparent Strain	Corrected Strain @ 5000 lb. Load
1145 lb.	threshold, fillet	30sec.	.00089in/in.	.00389in/in.
4530 lb.	1st in pin	19min. 00sec.	.00122in/in.	.00135in/in.

RAPID-LOADING TEST

Load	Crack	Total Time	Creep Corrected Apparent Strain	Corrected Strain @ 5000 lb. Load
1100 lb.	threshold, fillet	15sec.	.00086in/in.	.00390in/in.
3600 lb.	1st in pin	2 min. 45sec.	.0010 in/in.	.00139in/in.

Thus, for the first test with a total elapsed time of 19 minutes, against a total elapsed time of two minutes and 45 seconds in the second test, a difference of only forty microinches per inch exists after applying creep correction to obtain the apparent strains. This represents an error of less than three percent.

Strain gages were employed on the pin and on the cheek in the original and redesigned crank sections to obtain a comparison of results.

The following tabulated comparisons were acquired:

ORIGINAL CRANKSHAFT

Position	Stresscoat Indicated Strain	Strain gage Indicated Strain
Pin @ .42 in. from cheek	.00108	.00067
Cheek, area parallel to 180° position of crankpin fillet	.00115	.00078

REDESIGNED CRANKSHAFT

Position	Stresscoat Indicated Strain	Strain gage Indicated Strain
Pin @ .42 in. from cheek	.00115	.00084
Cheek, area parallel to 180° position of crankpin fillet	.00115	.00079

These values show a poor comparison, but when the strain ratios are computed the differences are well within the expected accuracy of Stresscoat.

ORIGINAL CRANKSHAFT

Strain Ratio	Stresscoat	Strain Gage
Cheek/pin	1.065	1.162

$$\text{Percent error} = \frac{1.162 - 1.065}{1.162} \times 100 = \frac{.097}{1.162} \times 100 = 8.3\%$$

REDESIGNED CRANKSHAFT

Strain Ratio	Stresscoat	Strain Gage
Cheek/pin	1.000	0.94

$$\text{Percent error} = \frac{1.00 - 0.94}{0.94} \times 100 = \frac{.06}{.94} \times 100 = 6.4\%$$

Since the level of strain at the location of the strain gage in the cheek is quite constant, the indicated strain by strain gage provides a calibration device for the lacquer. However, Stresscoat does provide strain ratios that compare favorably to the ratios obtained by the strain gages, and it is the ratios of strain that are interesting from a design point of view, since the maximum strain in the fillet remains at approximately the same value for the old and the new design.

It is interesting to note on Fig. 12 and 13 that the load vs strain plot for the cheeks on both the original and new crank sections practically coincide, while the slope of the load vs strain curve for the crankpin of the redesigned section has been decreased, indicating that the strain level has been raised in the new design.

Eleven tests employing Stresscoat, other than those in Figures 14 through 20, were made to obtain the correct experimental technique.

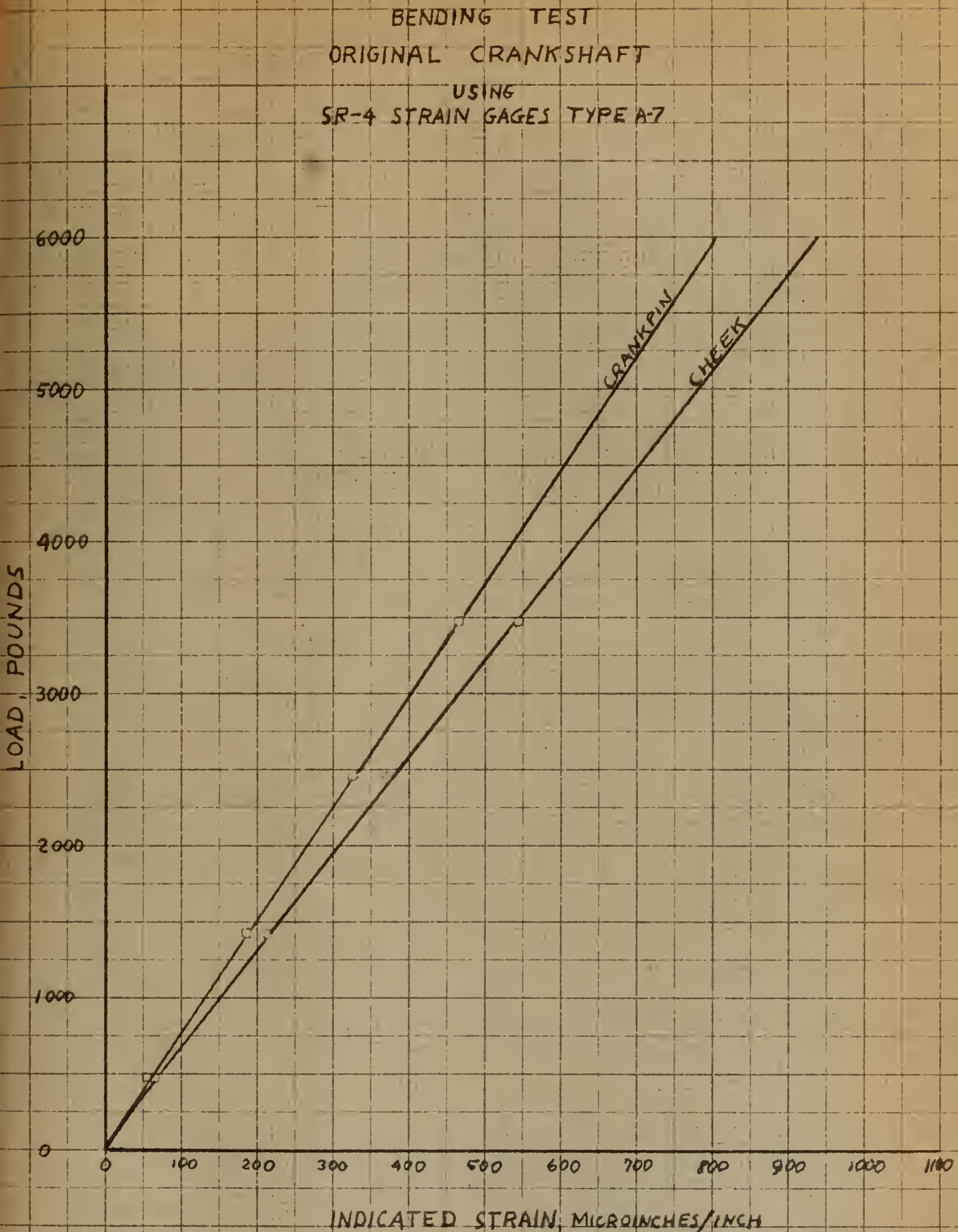

BENDING TEST
ORIGINAL CRANKSHAFT
USING
SR-4 STRAIN GAGES TYPE A-7
CRANKPIN
CHEEK
LOAD, POUNDS
0
1000
2000
3000
4000
5000
6000
0
100
200
300
400
500
600
700
800
900
1000
1100
INDICATED STRAIN, MICROINCHES/INCH

Fig. 12

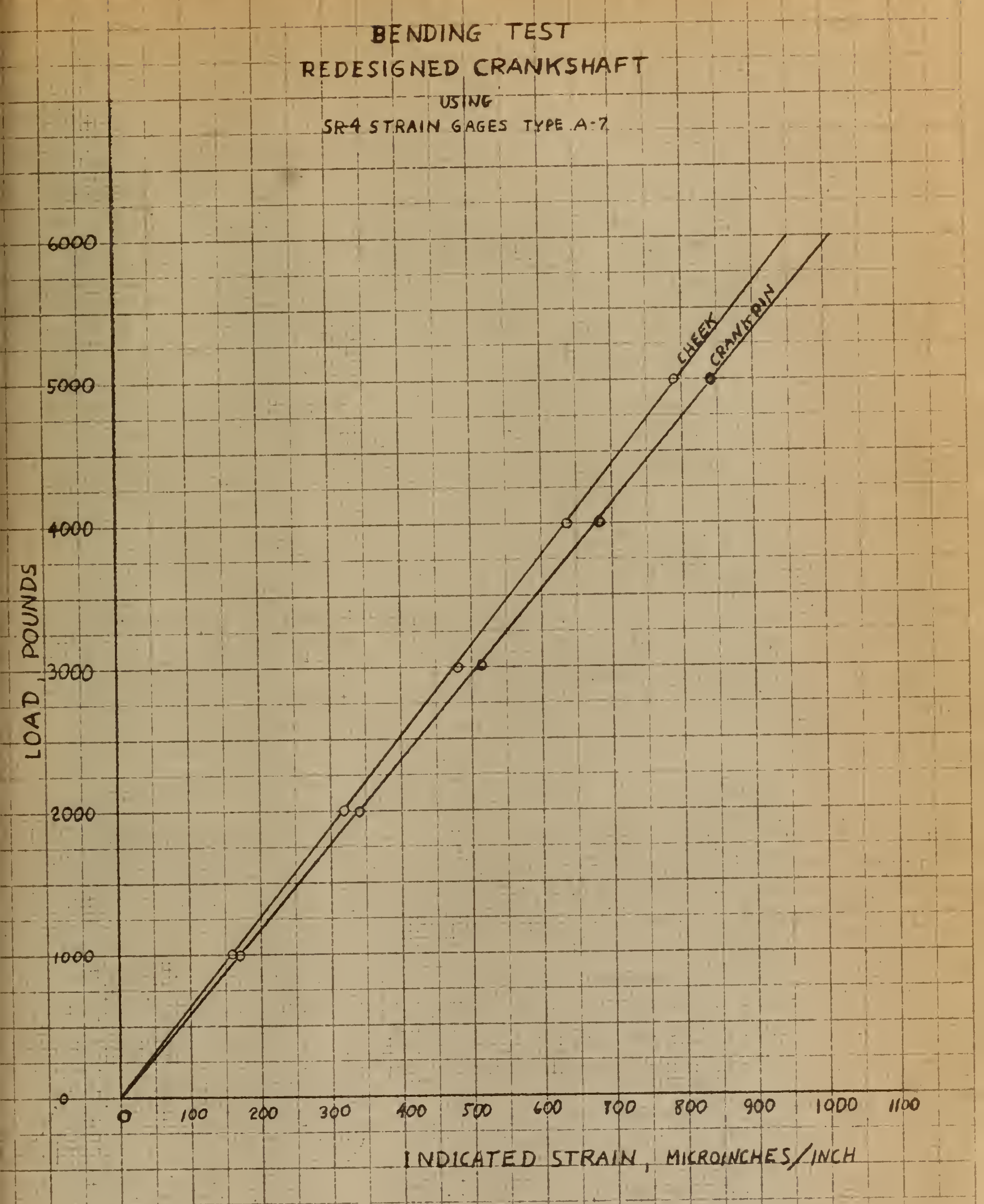
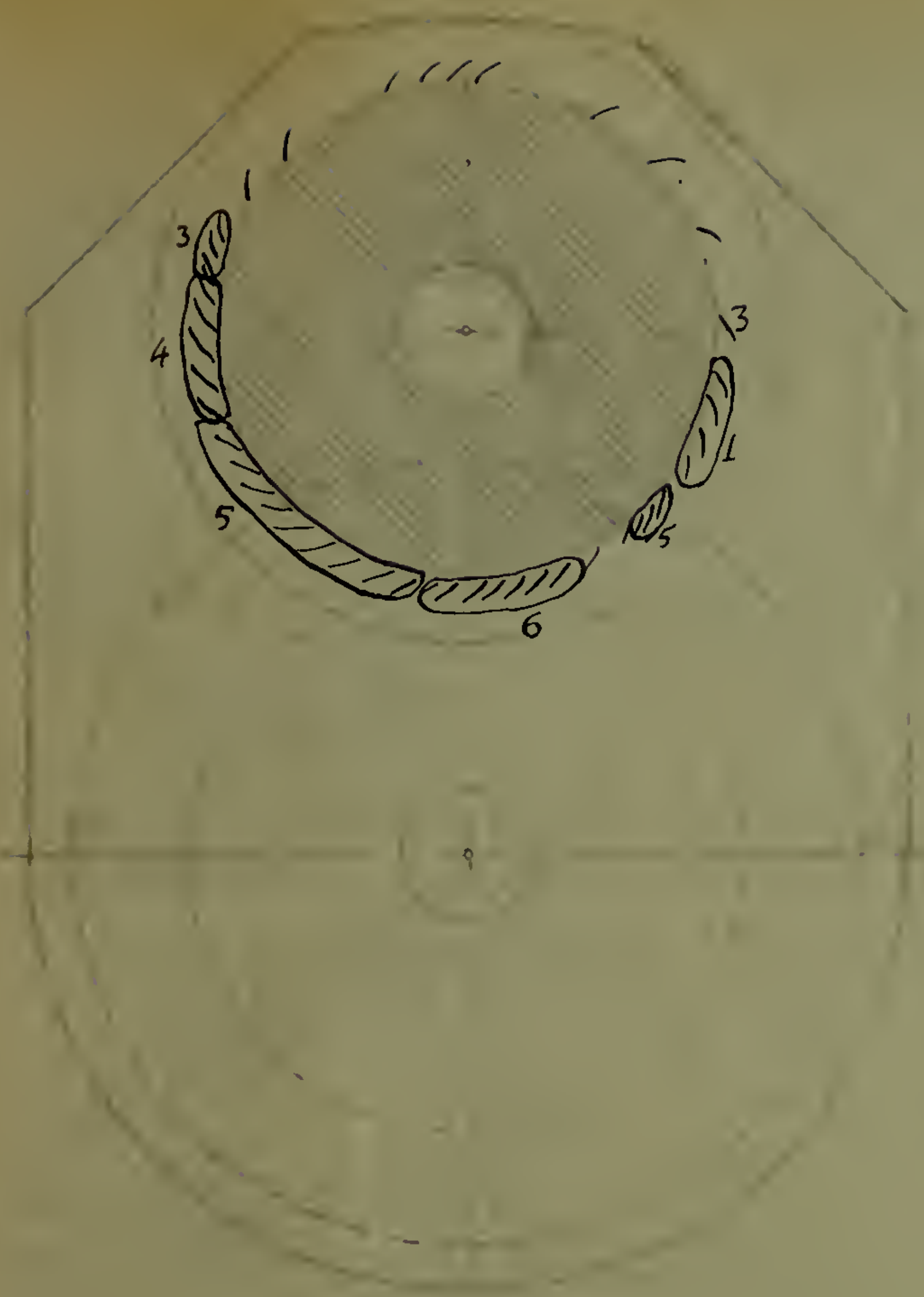
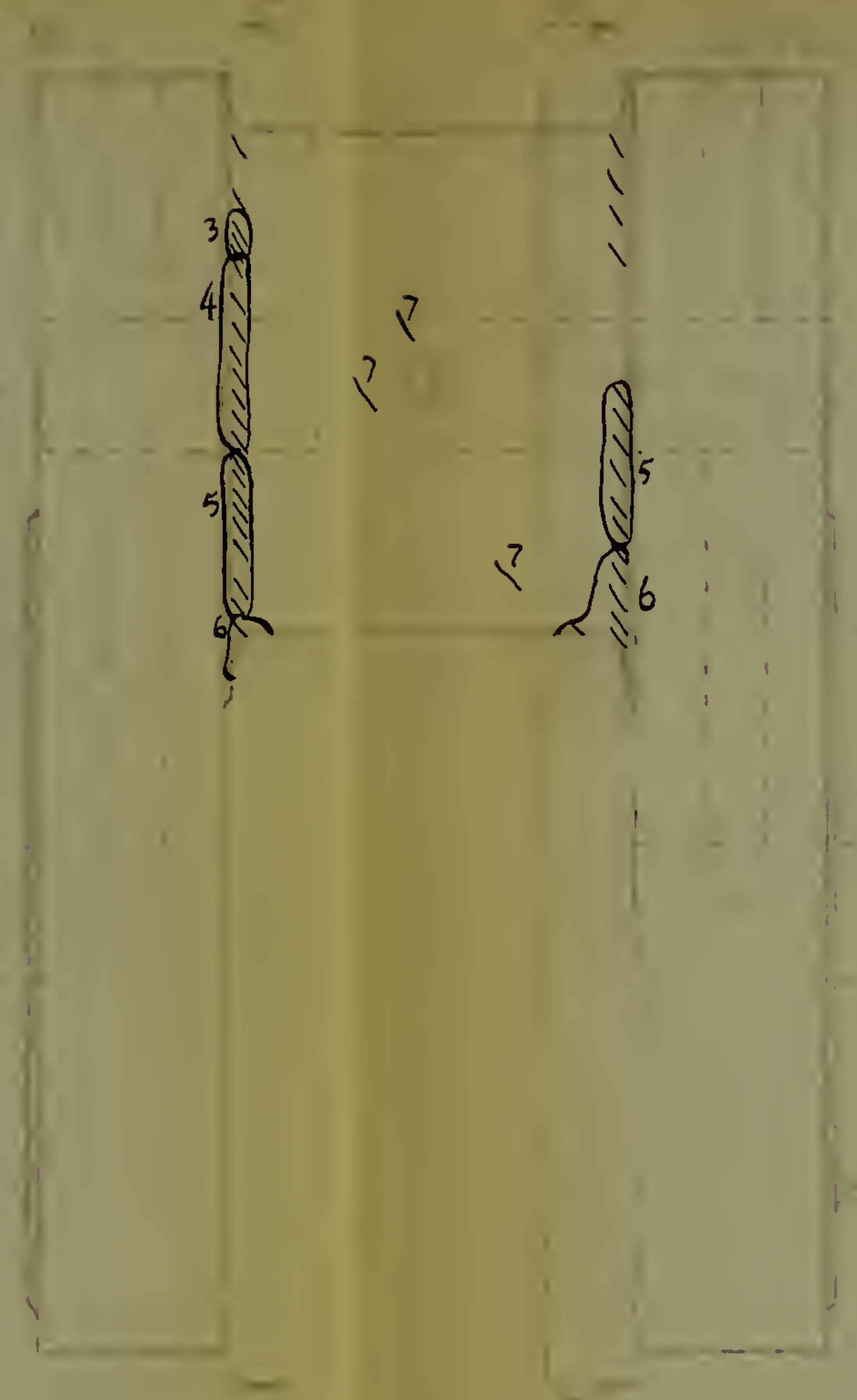
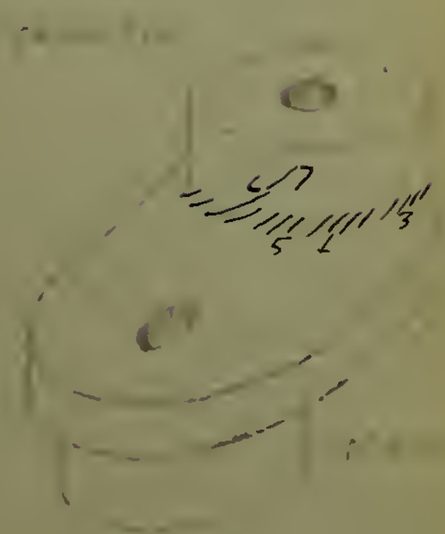

BENDING TEST
REDESIGNED CRANKSHAFT
USING
SR-4 STRAIN GAGES TYPE A-7
LOAD, POUNDS
6000
5000
4000
3000
2000
1000
0
0
100
200
300
400
500
600
700
800
900
1000
1100
CHEEK
CRANKPIN
INDICATED STRAIN, MICROINCHES/INCH

Fig. 13





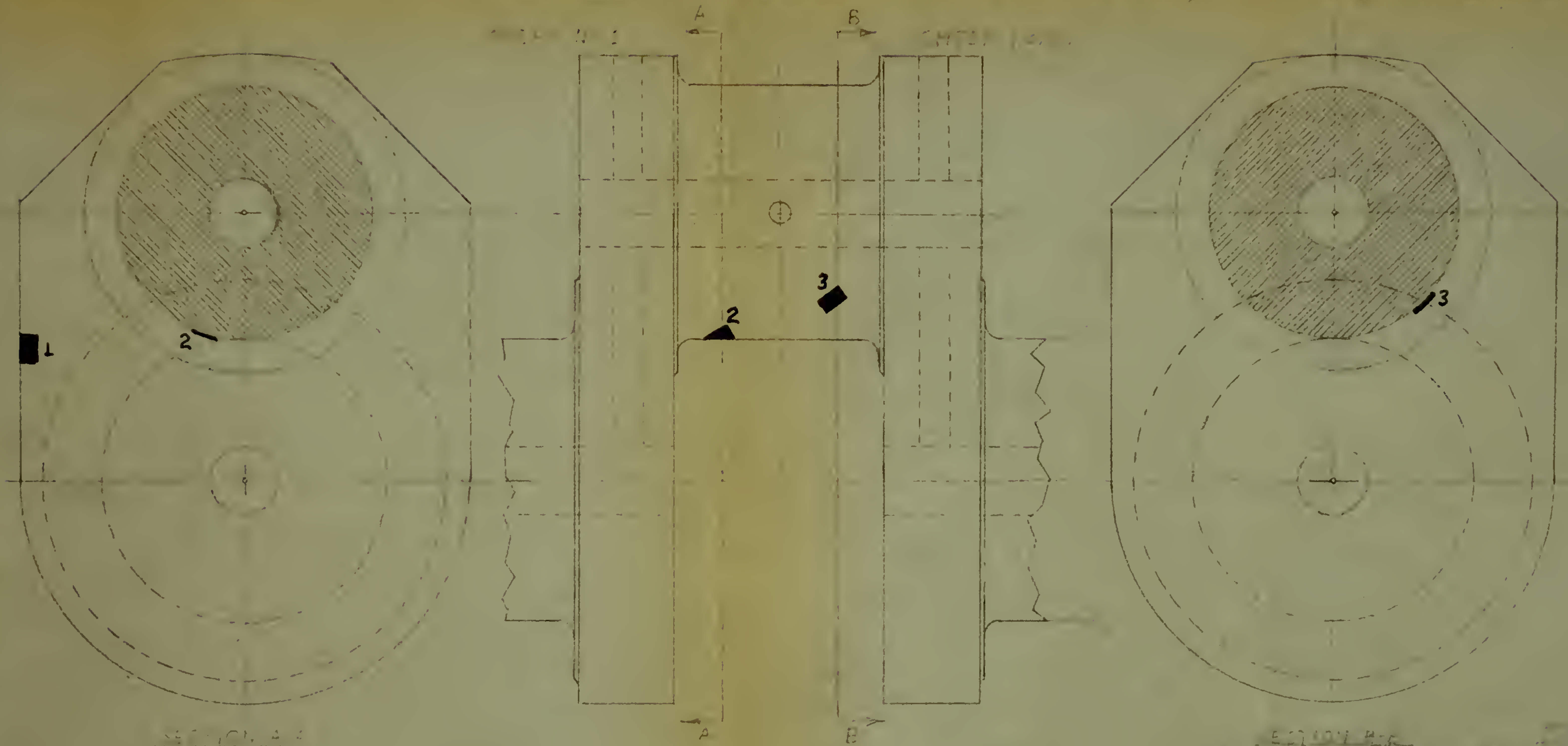


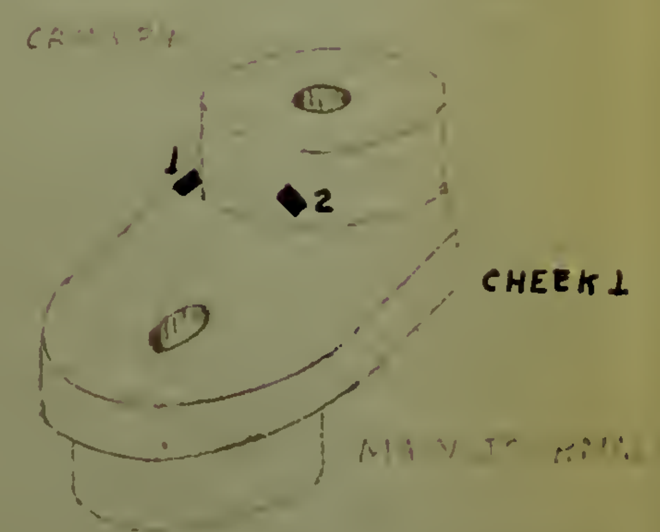
CHEEK 1

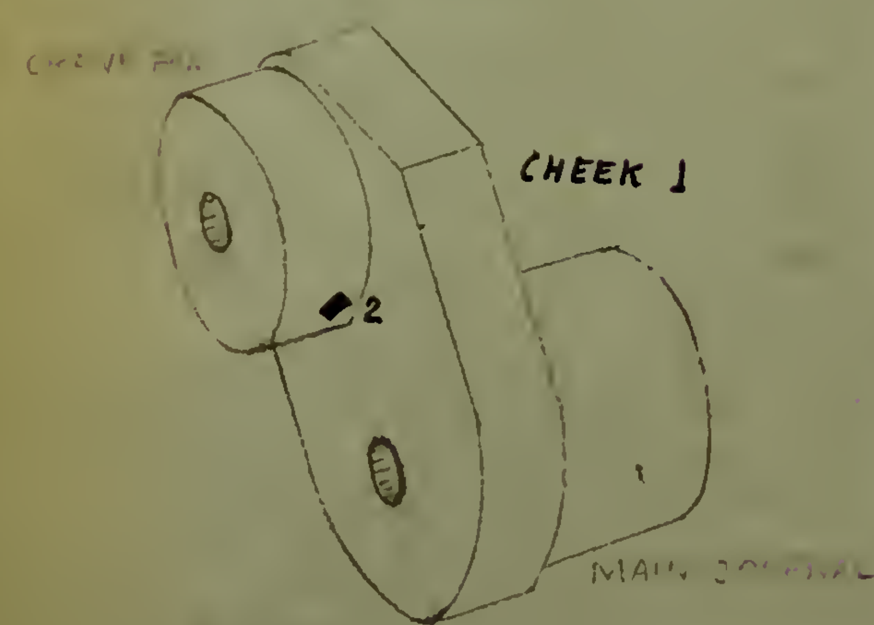
CHEEK 1

	LOAD IN POUNDS			
—	—	0m-00s	.00075	.00075
1	23,500	2m-15s		.00088
—	26,000	5m-30s		.00100
2	28,000	6m-15s		.00101
3	32,000	7m-00s		.00102
4	38,000	8m-15s		.00104
5	44,000	11m-45s		.00108
6	52,000	13m-30s		.00110
—	56,000	21m-15s		.00114
7	60,000	22m-00s	.00075	.00116

TEST: TORSION DATE: 5/5/55 TEMP: ____ °F wet; ____ °F dry



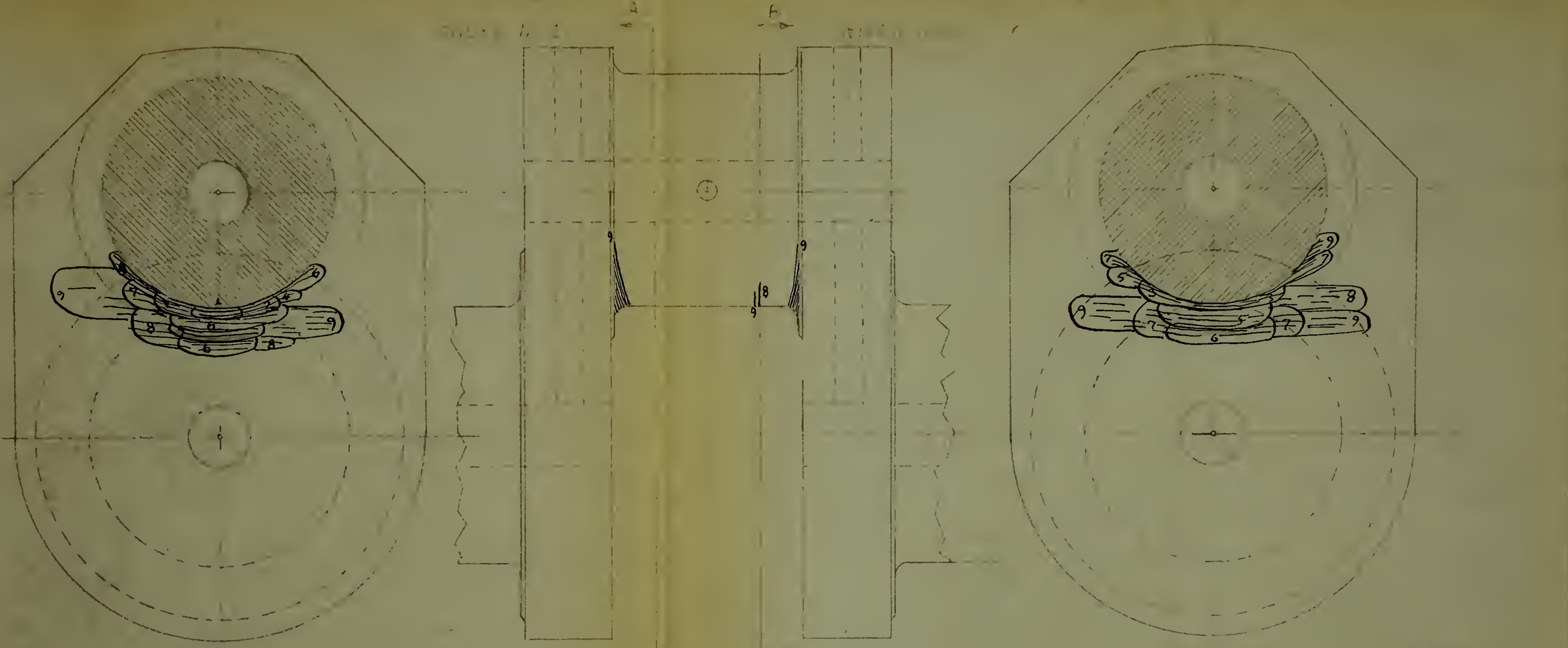




SR-4 Resistance wire strain gages type A-7 (1/4 in gage length) were attached at points where threshold cracks were formed from the brittle laquer test performed 5/2/55.

TORQUE	GAGE 1	GAGE 2	GAGE 3
IN-LBS	microin/in	microin/in	microin/in
2,000	20	48	30
18,350	284	428	315
35,500	407	829	612
53,750	605	1258	930

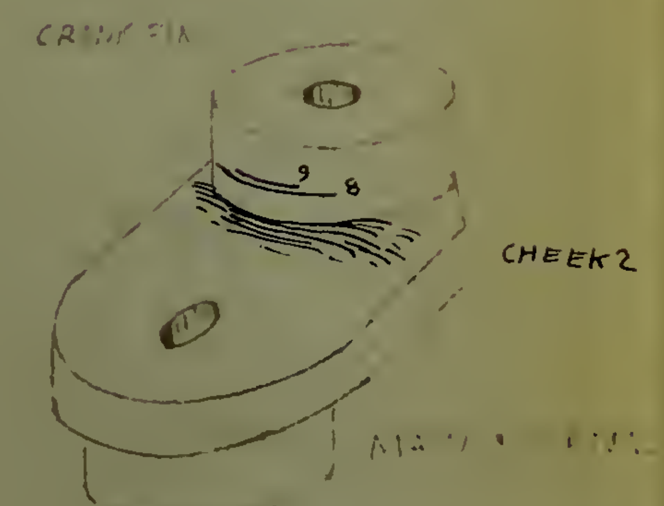
TEST: BENDING DATE: 4/28/55 TEMP: 58.0 77.0 1205



SECTION A-A

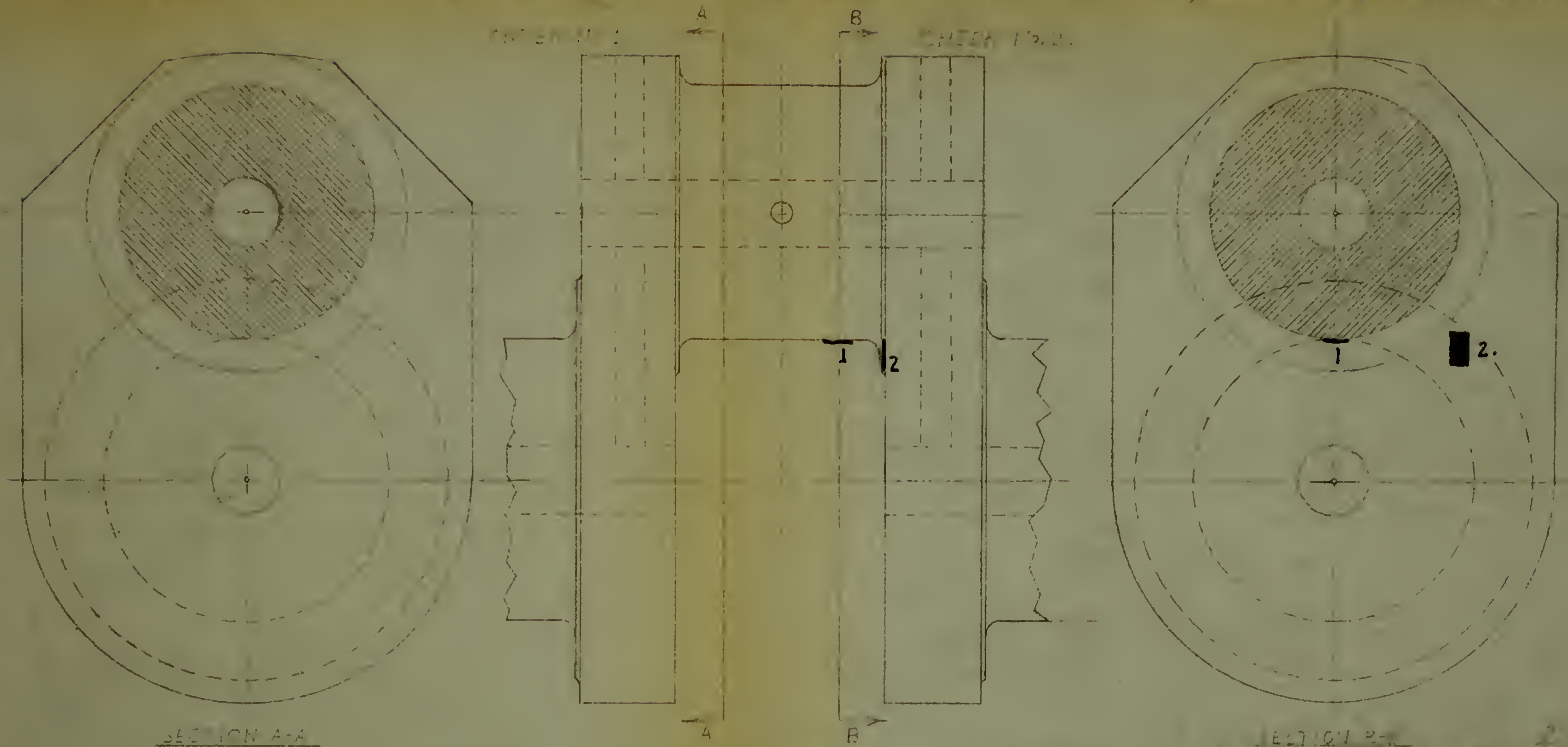
COMMENTS

Crack #8 in the pin occurred .4" from cheek, and crack #9 occurred .42" from cheek.



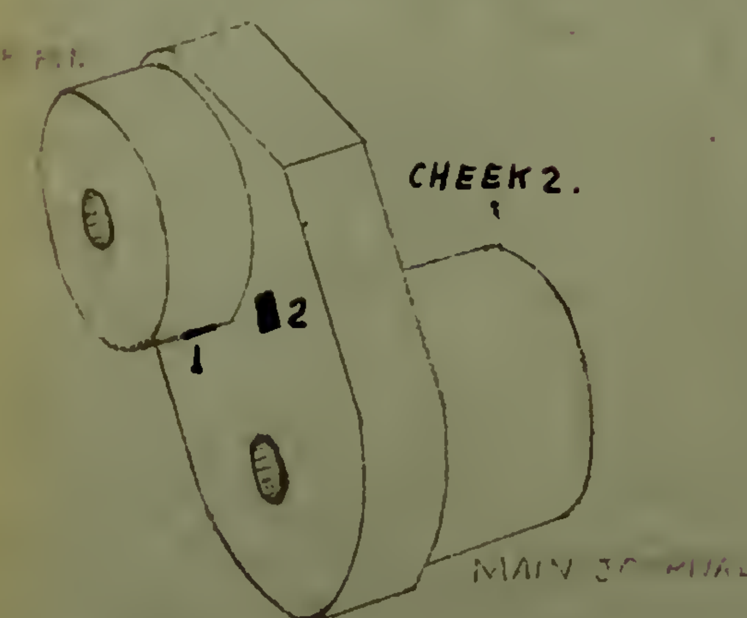
CRACKS	LOAD POUNDS	FT-LBS	TORQUE IN-LBS	TIME	SENSITIVITY IN/IN	CREEP
1	1200	300		1^m-18^s	.0008	.00095
2	1300	325		1^m-20		.00095
3	1500	375		5^m-45^s		.00107
4	2000	500		9^m-5^s		.00112
5	2500	625		14^m-0^s		.00118
6	3000	750		16^m-5^s	.0008	.00119
7	4000	1000		20^m-5^s		.00123
8	5000	1250		24^m-35^s		.00125
9	6000	1500		30^m-15^s	.0008	.00129

TEST: BENDING DATE: 5/7/55 TEMP: ______ °F ______ °F



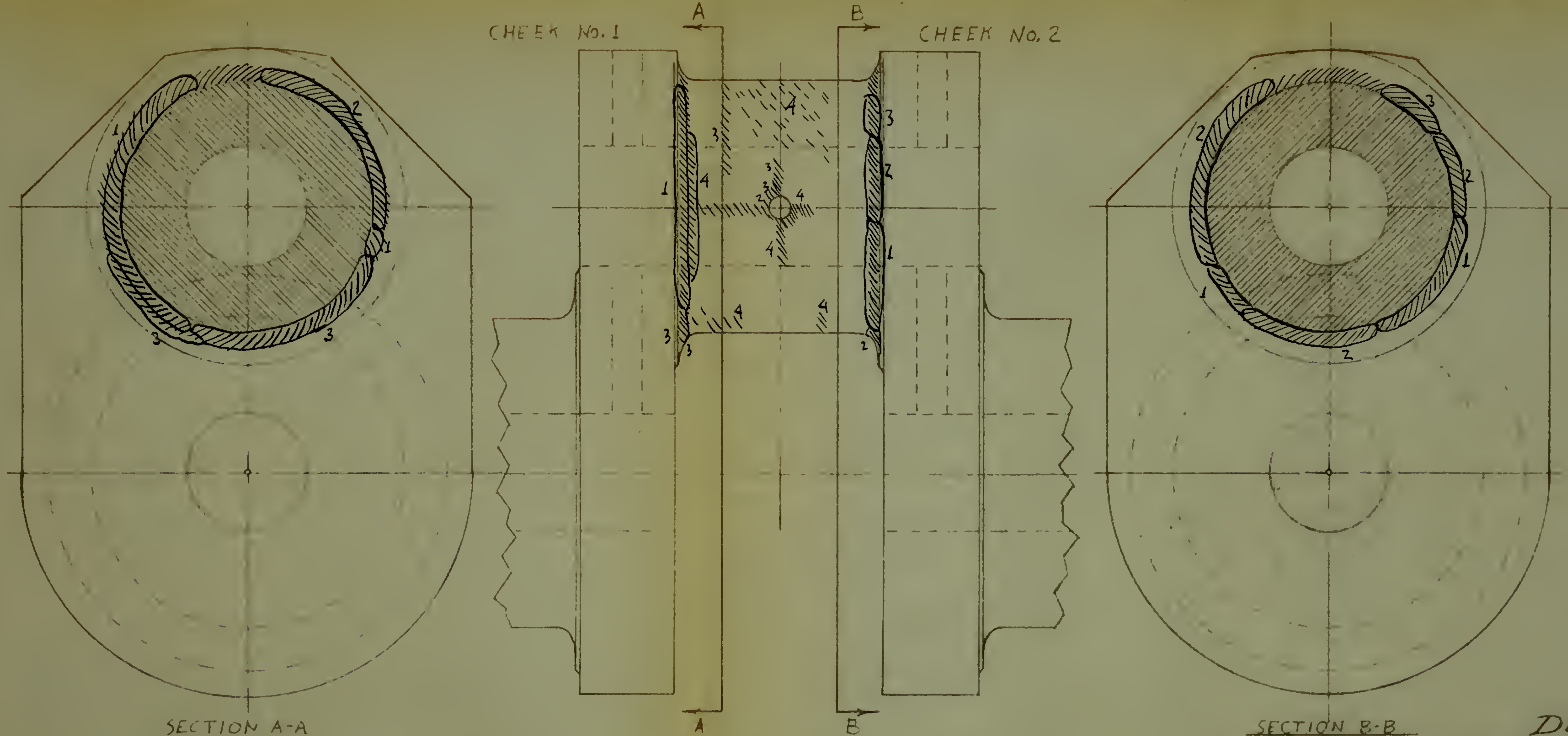
COMMENTS

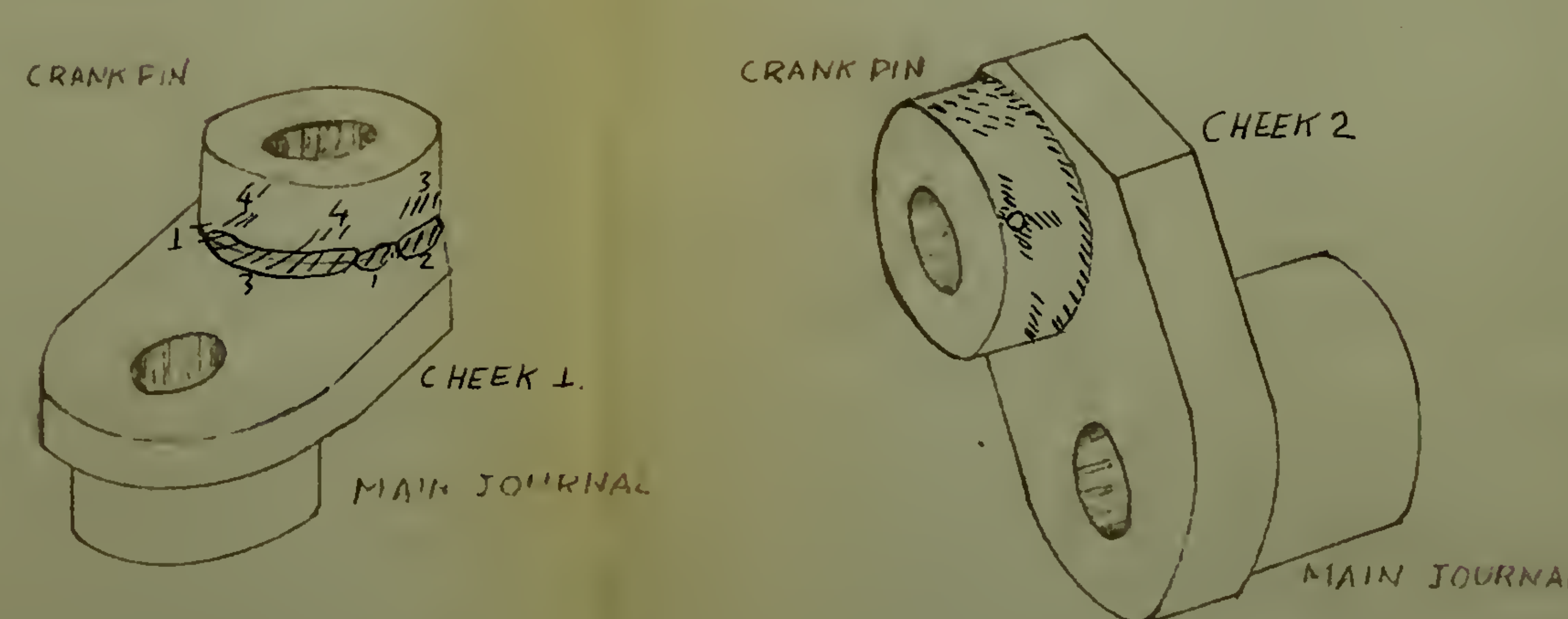
SR-4 Resistance wire strain gages (Type A-7, gage factor: 1.95) were attached at points indicated.



~~GAGES~~	LOAD Pounds	BEND MOM FT-lbs	~~TORQUE~~ in-lbs		SENSITIVITY in/in	CREEP	
			GAGE 1	GAGE 2			
			microin/in	microin/in			
	470	117.5	55	64			
	1420	355	184	212			
	2450	612.5	326	370			
	3475	869	466	543			

REDESIGNED CRANK SECTION TEST: TORSION DATE: 5/12/55 TEMP: 59.0 °F wet; 70.0 °F dry STRESSCOAT No. 1204





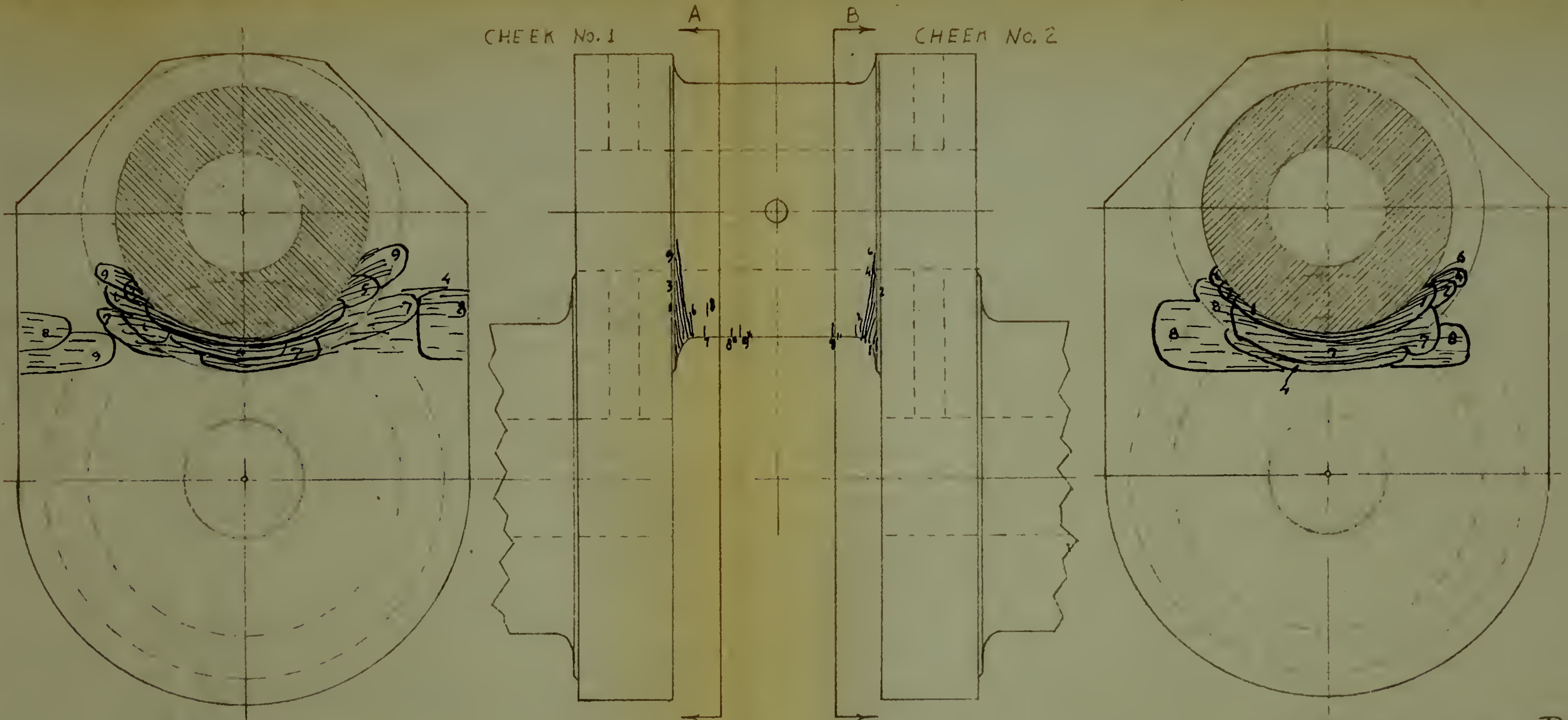
COMMENTS

The first cracks appeared over a much larger arc than in the original section. Cracks started moving out into that portion of the fillet having a large radius at a torque of 52,000 in-lbs.

Data:

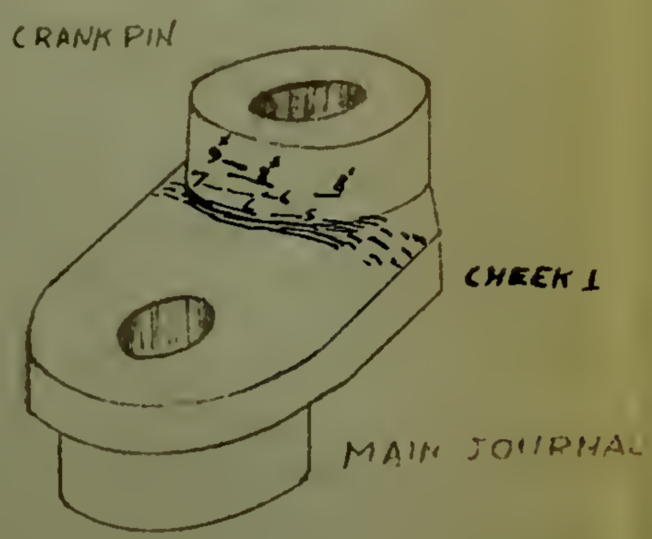
CRACKS	TORQUE IN-LBS	TIME	SENSITIVITY "/"	APPARENT STRAIN CORRECT. FOR CREEP			
—	—	0m-00s	.0008	.0008			
1	29,000	2m-15s	.0008	.00098			
—	36,000	7m-0s		.00109			
2	44,000	9m-15s		.00112			
3	52,000	13m-0s		.00116			
4	59,000	18m-30s	.0008	.00121			

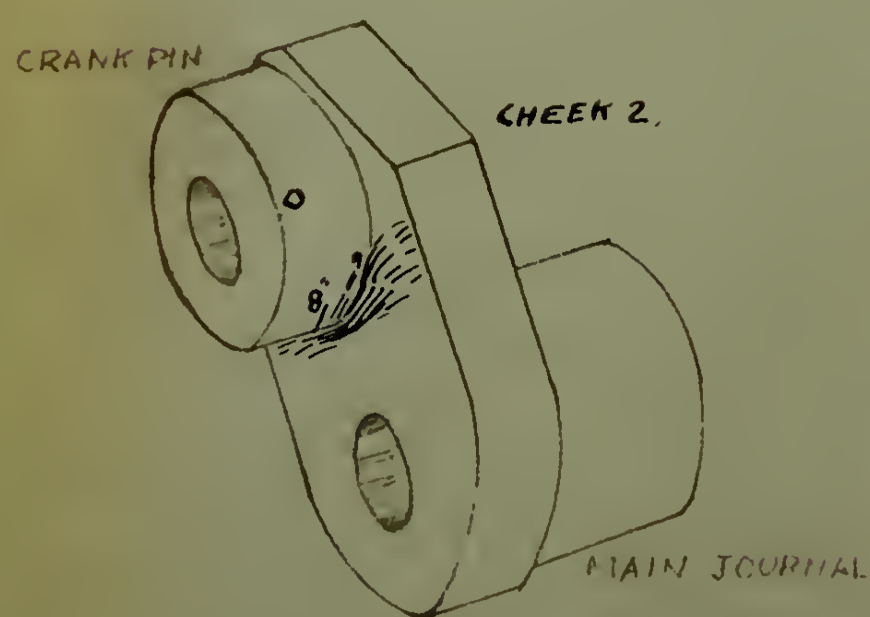
REDESIGNED CRANK SECTION TEST: BENDING DATE: 5/14/55 TEMP: 58.5 °F wet; 71.0 °F dry STRESSCOAT NO. 1204



SECTION A-A

SECTION B-B





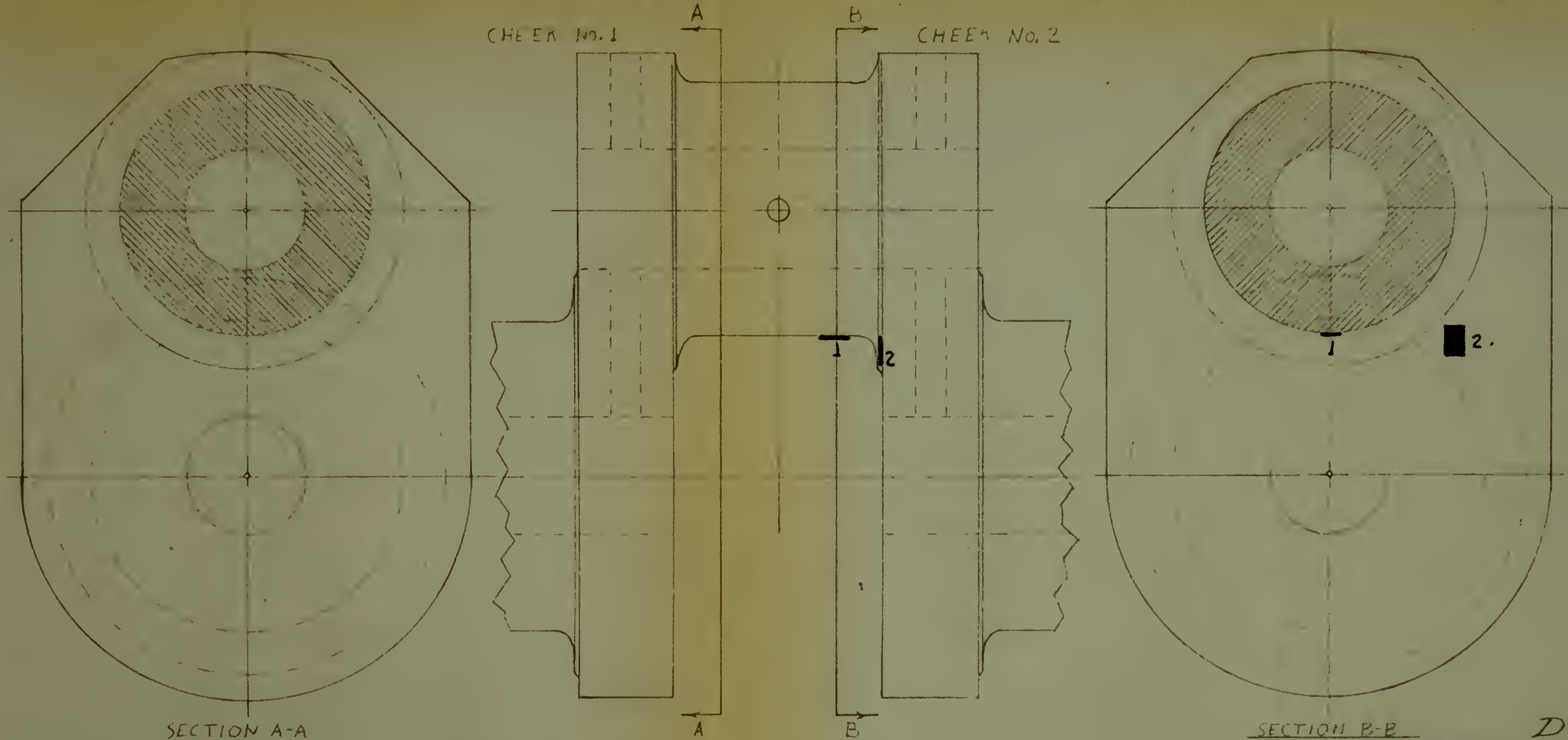
COMMENTS

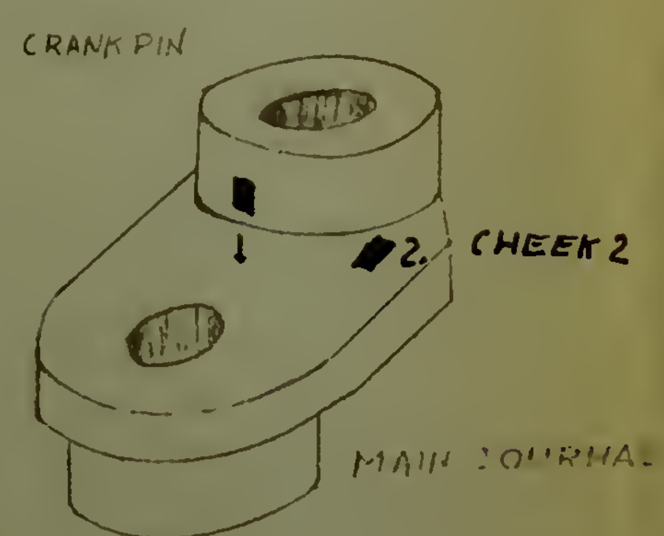
Crack 8*-0.5" out in pin from cheek #1, Crack #9*-0.58" out in pin from cheek #1 - both on center line. Crack 8' occurred at 45° & .42" out in pin from cheek 1. Crack 8" occurred .43" out in pin from cheek 2.

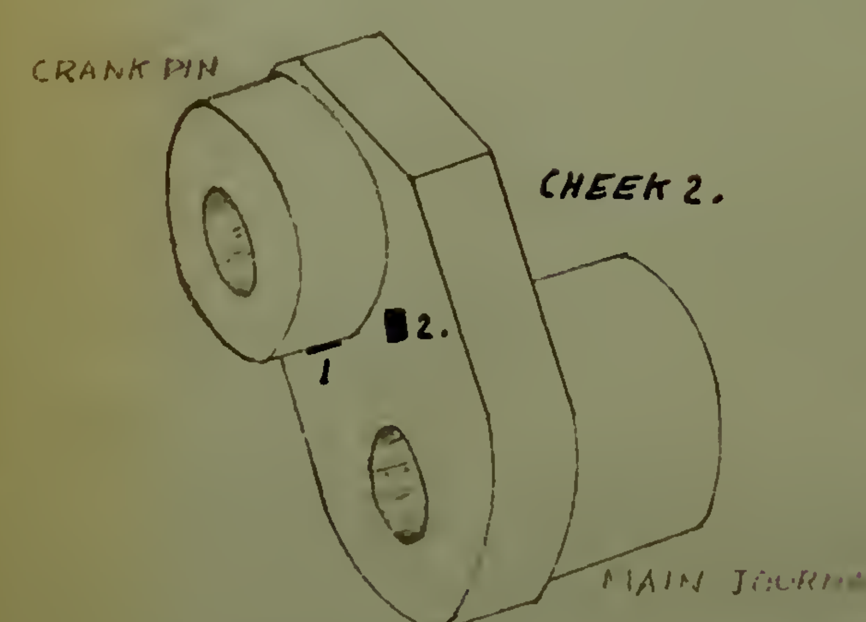
Data:

CRACK	LOAD LBS.	BEND. MOMENT FT-LB.	TIME	Sensitivity in/in.	Apparent Strain Creep Corrected		
—	—	—	0m-00s	.0008	.0008		
1	1145	286	0m-30s		.00089		
2	1525	381	3m-15s		.00101		
3	2035	509	5m-15s		.00106		
4	2420	605	7m-15s		.00110		
5	2845	711	12m-30s		.00116		
6	3525	881	16m-00s		.00119		
7	4530	1132.5	19m-00s		.00122		
8	5410	1352.5	22m-45s		.00124		
9	5880	1470	27m-15s	.0008	.00127		

CRANK SECTION TEST: BENDING DATE: 5/20/55 TEMP: ______ °F wet, ______ °F dry STRESS...







COMMENTS

SR-4 Resistance wire strain gages (Type A-7, Gage factor: 1.95) were attached at points indicated.

Data:

LOAD LBS.	BEND. MOM. FT-LBS	GAGE 2 microin./in.	GAGE 1 microin./in.
1,000	250	60	71
2,000	500	318	342
3,000	750	480	514
4,000	1000	635	682
5,000	1250	789	840

CHAPTER VII

CONCLUSION

1. Findings

The following items were determined by this work:

(a) The failure in the original crankshaft is a bending failure.

(b) The redesigning of the crankshaft proved inconclusive with respect to improving the shaft strainwise.

(c) The redesigning of the shaft did materially aid in providing a better balance for the engine.

(d) Since the engine is in better balance, the maximum enertia forces acting on the shaft are decreased by approximately 20 percent. (With the external shoulders of the cheeks removed, this will decrease the enertia forces by 35 percent.) The reduction of unbalanced enertia forces acting on the crankshaft, will result in lower values of maximum stress.

2. Recommendations

(a) That the lightening holes be bored to diameters of 1.274 inches for the crankpin and 1.424 inches for the main journals.

(b) That the segments be removed from the cheeks.

(c) That the junction of the fillets with the cheeks be faired in more smoothly because these internal corners have been found to nucliate cracks in the brittle lacquer.

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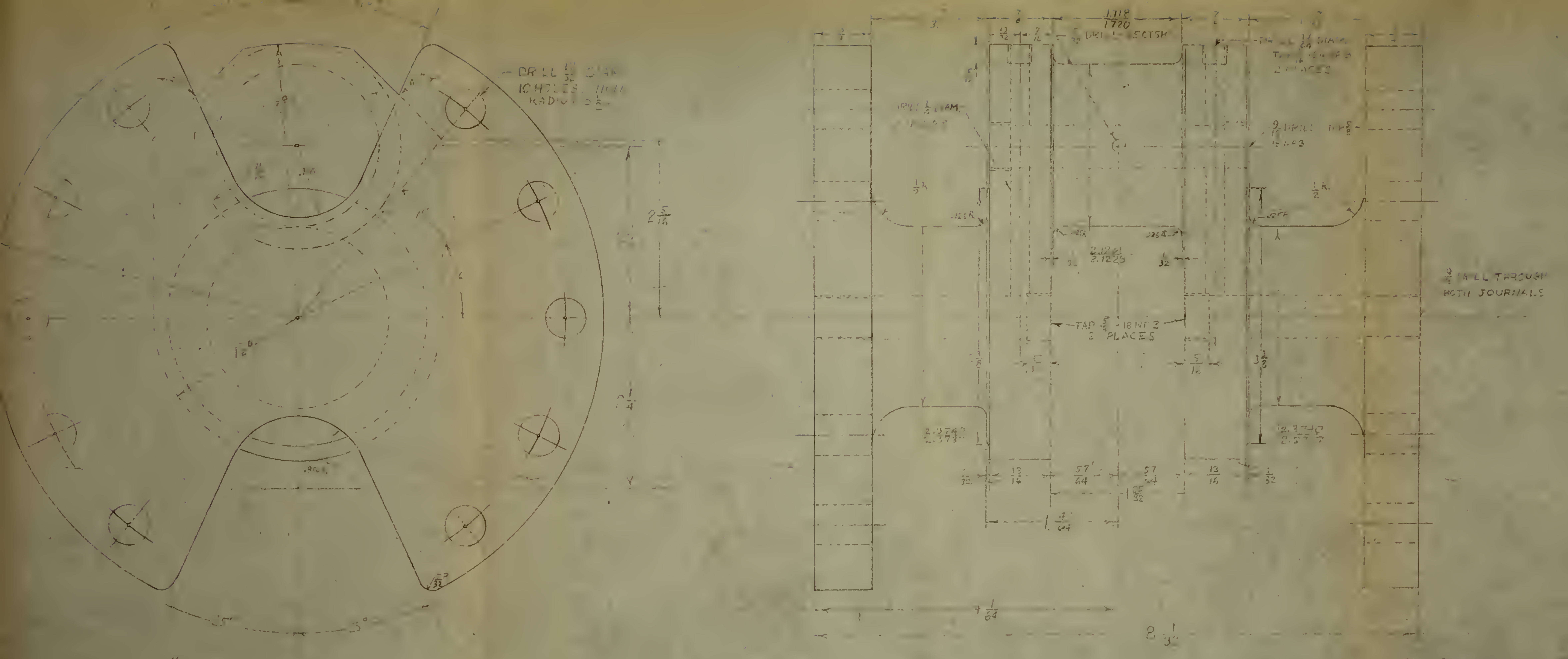

Fig. 21

U.S. NAVAL POSTGRADUATE SCHOOL
MECHANICAL ENGINEERING DEPARTMENT

EXPERIMENTAL CRANKSHAFT No. 1.
BY G.B. LINDGREN

Fig. 22

U.S. NAVAL POSTGRADUATE SCHOOL
MECHANICAL ENGINEERING DEPARTMENT

EXPERIMENTAL CRANKSHAFT (COMPOSITE)

TOLERANCES UNLESS OTHERWISE NOTED:
FRACTIONS ± .015 — DECIMAL ± .005 — ANGULAR ± 1/2 DEG

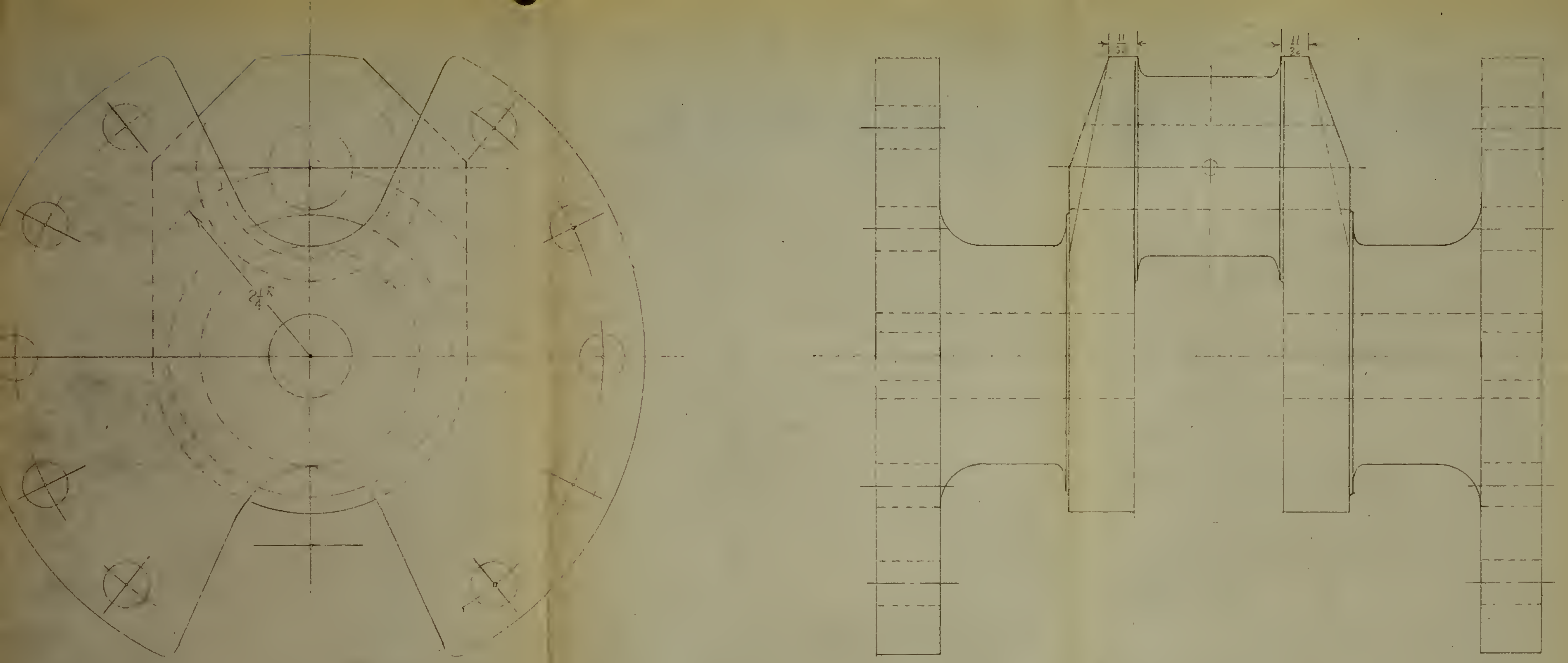

Fig. 23

U.S. NAVAL POSTGRADUATE SCHOOL
MECHANICAL ENGINEERING DEPARTMENT
EXPERIMENTAL CRANKSHAFT (COMPLETE)
TOLERANCES UNLESS OTHERWISE NOTED
FRACTIONAL ± .015 — DECIMAL ± .005 — ANGULAR ± $\frac{1}{2}$ DEG.

Thesis
L648
28462
Lindgren
Investigation of failures of a high-speed crankshaft.